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GERMAN SUBMARINE ENGINES.

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In view of the prominence attained by the Diesel Engine in Submarine practice, and of its more recent application to an increasing degree in commercial practice, the following brief description of a German Submarine engine obtained from inspection of a number of these vessels is thought will be of interest to the service. The German answer to many questions which have puzzled our designers and in many cases which are still open questions may be found in these engines. Prominent points in this answer are: The four-stroke cycle; air injection; air starting and reversing; high revolutions and piston speed; oil-cooled pistons; extreme rigidity; stroke-bore ratio close to 1.0 or less; and the application of six cylinders to powers up to 1,750 B.H.P. The following table shows the general characteristics of the machinery installation of thirty-six submarines inspected.

Boat.	Make of Engine.	Number in Boat.	Cycle.	Number of Cylinders.	Horsepower Engine.	Revolutions Per Minute.	Air. Starting.	Air. Reversing.	Year.
U-19	M. A. N.	2	4	6	850	450	Yes	Yes	1912
U-53	M. A. N.	2	4	6	1,200	450	Yes	Yes	.....
U-55	M. A. N.	2	4	6	1,200	450	Yes	Yes	1916
U-67	Krupp	2	2	6	(1)	(1)	Yes	Yes	.....
U-79	Vulcan	2	4	6	500	450	Yes	No	.....
U-101	M. A. N.	2	4	6	1,200	450	Yes	Yes	.....
U-108	M. A. N.	2	4	6	1,200	450	Yes	Yes	.....
U-112	Krupp	2	2	6	1,600	350	Yes	Yes	1917
U-114	Krupp	2	2	6	(2)	(1)	Yes	Yes	1917
U-117	M. A. N.	2	4	6	1,200	(1)	Yes	Yes	.....
U-119	M. A. N.	2	4	6	1,200	450	Yes	Yes	1917
U-120	M. A. N.	2	4	6	1,200	450	Yes	Yes	.....
U-122	M. A. N.	2	4	6	1,200	450	Yes	Yes	.....
U-124	(3)	2	4	6	1,200	450	Yes	Yes	.....
U-125	.....	2	4	6	1,200	450	Yes	Yes	1917
U-135	.....	2	4	6	1,750	380	Yes	Yes	1917
	In addition, this vessel has charging engines as follows:								
.....	M. A. N.	2	4	...	450	400	Yes	No	1917
U-139	Krupp	2	2	6	1,600	350	Yes	Yes	1915
	Designed for one charging engine, but not installed:								
U-141	M. A N.	2	4	6	1,750	380	Yes	Yes	1917
	In addition, this vessel has one charging engine, as follows:								
.....	M. A. N.	1	4	6	450	400	Yes	No	1917
U-146	M. A. N.	2	4	6	1,000	450	Yes	Yes	1915
U-153	Krupp	2	4	6	(2)	.....	Yes	No	1916
U-160	M. A. N.	2	4	6	1,200	450	Yes	Yes	.....
U-164	M. A. N.	2	4	6	1,200	450	Yes	Yes	1917
UB-99	Vulcan	2	4	6	550	450	Yes	Yes	.....
UB-112	M. A. N.	2	4	6	550	450	Yes	Yes	1917
UB-114	M. A. N.	2	4	6	550	450	Yes	Yes	1917
UB-117	Vulcan	2	4	6	550	450	Yes	Yes	.....
UB-122	Korting	2	4	6	530	450	Yes	Yes	.....
UB-131	Benz	2	4	6	530	450	Yes	.....	1918
UB-149	Korting	2	4	6	530	430	Yes	.....	1918
UC-94	M. A. N.	2	4	6	300	450	Yes	No	.....
UC-98	M. A. N.	2	4	6	300	450	Yes	No	.....
UC-101	M. A. N.	2	4	6	300	450	Yes	No	1918
UC-103	M. A. N.	2	4	6	300	450	Yes	No	1918
UC-104	M. A. N.	2	4	6	300	450	Yes	No	1918
UC-105	M. A. N.	2	4	6	300	450	Yes	No	1918
UA-	Krupp	2	4	6	(1)	.....	Yes	No	1913

(1)—Not known (2)—About 1,000 (3)—Blohn-Voss

The above table shows the engine installation of thirty-six boats taken at random; of these five are equipped with two-cycle engines. The two-cycle engines inspected were all of the Germania Krupp type and similar, as far as could be seen, to the Krupp design before the war. The engines of the four-cycle type were manufactured in almost all cases by the M.A.N Co. The few other makes found so resembled the M.A.N. type as to indicate their manufacture from the M.A.N. design.

The prevalency of the M.A.N. four-cycle engine proved beyond doubt the superiority of this type in Germany. For this reason that type only was examined in detail and furnishes the material for as much detailed information as it was possible to collect for this report. The inspection was concentrated on the engines of *U-164*, which are six-cylinder, four-cycle, air starting and reversing, 1,200 horsepower, at 450 revolutions per minute.

Sizes.—The M.A.N. engine was found mainly in four sizes, namely:

300 horsepower at 450 revolutions per minute,—stroke, 10.937-inch bore.

550 horsepower at 450 revolutions per minute,—stroke, 12.792-inch bore.

1,200 horsepower at 450 revolutions per minute, 17-inch stroke, 17.750-inch bore.

1,750 horsepower at 380 revolutions per minute, 530-mm. stroke, 530-mm. bore.

The exceptions to the above four standard types are the 850-horsepower M.A.N. four-cycle engine of an old model, 1912, and the charging engines on the submarine cruisers *U-135* and *U-141*. These charging engines are the M.A.N. design, 450 horsepower at 400 revolutions per minute, air starting.

All the above types are made in six-cylinder units and are air starting; all but the 300-horsepower type are reversing.

Material.—A shortage of copper is noticeable in the con-

struction of these engines or rather a progressive shortage, for some of the earlier Krupp engines are made of bronze. Steel is used to a very large extent; the cylinders, heads, housings, bedplates and piping, together with other parts which are found in general practice, are of this material. The bedplate and housing should be especially noted in this connection since it is rather a complicated casting with parts as thin as one-half inch or less. Engine parts were not painted, but apparently all unfinished castings were painted with an anticorrosive wash or iron filler paint.

Engine-Room Arrangement.—The general arrangement of the engines and auxiliaries is similar to our practice. The two engines are placed abreast and turn outboard going ahead. The attached lubricating pumps are driven from the vertical shaft at the after end of the engines. The attached circulating pumps are driven off the crankshaft at the forward end of the engines. The two auxiliary lubricating pumps are forward and outboard of the engines, above the floor plates and are motor driven on horizontal shafts. The two auxiliary circulating water pumps are at the forward end of the engine room; the motors are above the floor plates and the pumps below, the shafts being vertical.

The control of the engine is at the forward, inboard side of each engine.

Spares and tools are stowed in metal lockers, built in, outboard of the engines, forward of the engines, under the floor plates, between the engines and in racks overhead and outboard on the hull.

Except in the boats larger than the 800-ton type the spares consist of small engine and auxiliary parts, such as valves, etc. Only on the large boats are cylinder heads or pistons found.

Tools, both standard and special, are neatly stowed in racks with a silhouette of each tool painted on the rack to insure proper stowage. Special tools are all numbered for the purpose intended.

Working room around the engines is limited, particularly abaft and outboard of the engines; however, the engines are generally accessible. Disassembling gear is provided overhead; athwartship tracks and chain falls; but apparently the engines are found to require little or no major overhaul at sea. The floor plates are pressed, bumped steel, are very strong and accurately fitted. They are secured to the angle irons by means of dogs operated by flush head bolts, slotted for a special wrench or large screw driver.

Cylinder Heads.—Cylinder heads are made of cast steel, machine-finished outside. The top and bottom surfaces are practically flat and the horizontal section square with rounded corners. Adjacent heads are held together by means of steel tie plates.

There are six valve openings in the head. This arrangement is varied in the 300-horsepower engines on account of the single-spray valve.

Passages, open to the starting valves, are provided, fore and aft, through the inboard part of the head. These starting air passages are connected between cylinders by jumpers. The whole serves as a starting air manifold.

The arrangement of valve openings and cooling spaces is very symmetrical, allowing of equal expansion under running heat.

Castings are made in rights and lefts in order that adjacent heads may be placed with inlet or exhaust passages adjacent.

The cylinder-head joint is of the usual spigot type, with copper gasket similar to our practice.

Cylinder heads are secured by means of eight steel bolts, 2⅛ inches in diameter.

The indicator cock connection is through the inboard side of the head.

Cylinders.—Engine cylinders are of cast steel with separate liners. The jacket and skirt which form the upper half of

the housing are cast integrally. Webs and flanges are provided on the forward and after sides for bolting to adjacent cylinders and at the bottom for bolting to bedplate flange joints. These are metal-to-metal and the whole construction is very substantial and rigid. On the 300-horsepower engines the working cylinders are cast in pairs. Handholes are provided in the water jackets for cleaning. The cylinder liner is apparently made of close-grained cast iron. It is secured at the top where the head presses it against a shoulder in the cylinder casting; a stuffing box is provided at the bottom of the cylinder jacket to provide for lateral expansion of the liner and at the same time insured watertightness of the water space between jacket and liner. The stuffing box is secured by tap bolts, wired.

Bedplate.—The bedplate is of channel construction, generally similar to our practice, except that the sides are carried up much higher. There is no separate housing provided; the skirts or lower parts of the cylinders and the high sides of the bedplate form this part. Abreast each main bearing, inboard and outboard, there is built up a girder construction connecting the bedplates and cylinders, which gives great vertical strength and athwartship rigidity. This construction, together with the bolted connection between cylinders at the top and the cylinder-head tie-plates, makes the whole engine extremely rigid and free from vibration when running. The bedplates for the 1,200-horsepower engines are made up in three sections with bolted metal-to-metal joints.

Pistons and Wrist Pins.—The working pistons are of the trunk type, and appeared to be made of fine close-grained cast iron. Each piston is provided with five rings at the top, the lower one being a wiper ring, and one wiper ring about 5 inches from the bottom. The piston head is dished, forming a combustion chamber of cylindrical shape with flat top and dished bottom. Piston rings are lap-jointed, and are fitted with dowel pins to prevent rotation. Piston clearance on the

1,200-horsepower engines as taken with feelers was found to be roughly 0.016 inch at the bottom, and 0.062 inch at the top.

The wrist pin is of case-hardened forged steel. It is straight, not tapered. The wrist pin is secured by two large bolts passing through the bosses and piston walls. The securing bolts are in turn secured by set screws.

Valves.—All principal valves are placed vertically in the cylinder heads. The inlet and exhaust are poppet valves of the usual type, the exhaust valve being water-cooled (by a connection to the circulating water system). The inlet valves are of cast steel in one piece. The exhaust valve stems are of cast steel, on which are welded cast-iron heads. A cooling chamber is cored in the exhaust-valve head. Cooling water is forced in through a small tube in the stem and discharged out through the annular space around the supply tube. The lift of the inlet and exhaust valves is about 1.5 inches.

Air-starting valves and relief valves are very similar to our practice. The relief valves are provided with air plungers for positive lifting when reversing the engines.

The spray valves are provided in pairs to each cylinder, except on the 300-horsepower engine. The lift of the valve spindle is controlled to choke the spray air at low speeds. Each spray valve is provided with a nozzle plate drilled with six 2.4-millimeter holes. The internal construction of the spray valve, the atomizer, etc., is according to general practice. One special feature is the actuating connection between rocker arm and the spray-valve spindles. One rocker arm, which is forked on the valve end and carries a tappet plate on trunnions, actuates both valve spindles. Instead of the usual practice of lifting the spindle by direct contact between the tappet and a fixed shoulder on the spindle, in this engine a spring is interposed.

The spray-valve cam begins to take on the roller about 30 degrees before top firing center, but the valve itself does not open until about 8 degrees to 10 degrees before center.

Valve Gear.—The cam shaft is located on the inboard side of the engine at the level of the heads and is driven by worm and gear from a vertical shaft at the after end. The vertical shaft is driven by worm and gear from the crankshaft. The 2:1 gear ratio is obtained in the gears between the vertical shaft and cam shaft. The cam shaft turns inboard going ahead. The cams are solid and keyed on the shaft, except the spray cam, which is an adjustable insert on a carrier keyed to the shaft. Two cams are provided for each valve—one ahead and one reverse. The cam-shaft bearings are carried in supports which bolt rigidly to the flanges where the cylinders are joined together. The same supports also carry the bearings for the rocker shaft, which is located directly over the cylinders. The rocker shaft forms the pivot for all the valve rockers, or rather carries eccentric sleeves on which all the valve rockers pivot. The inlet and exhaust rockers are carried on fixed eccentric sleeves on the rocker shaft so that by turning the shaft the valve rockers are lifted. The spray and air-starting rockers are carried on loose eccentric sleeves on the rocker shaft and are thrown in or out by means of the starting levers. All valve rocker arms are held with their tappets against their respective valve stems, the play always being at the roller end where the sloping surface of the cam will start the lifting motion without striking.

Crank Shaft.—The crank shaft is a hollow steel forging in one piece, including the air-compressor cranks. The main cranks are set at 120 degree angles, Numbers 1 and 6, Numbers 2 and 5, and Numbers 3 and 4 being at the same angle. The air-compressor cranks are at 180 degrees and in phase with cranks Numbers 1 and 6. The sequence of cranks is 1, 4, 2, 6, 3, 5 for the starboard engine and 1, 5, 3, 6, 2, 4 for the port engine (Number 1 cylinder is considered forward). The crank shafts are apparently interchangeable. No thrust bearing appears to be fitted, but the end clearances of all crank webs is very small, probably to provide for that purpose. No balance weights are fitted to the crank webs.

Air Compressors.—Four-stage air compressors are provided on all the types noted except the 300 horsepower, which has a three-stage compressor. The cylinders are made in twin castings and are tandem. Air-compressor cranks are set opposite. The lower pistons are double acting, the first stage compression taking place in both cylinders on the down stroke. The lower pistons are of the trunk type, and second-stage compression is obtained in the annular space around the lower pistons. The third and fourth stage compressions are obtained in the two upper cylinders, respectively, single acting. Air discharge to coolers is screened, but not cooled. Air compressor valves are apparently very reliable and require little overhaul. Engine air compressors are used to charge starting bottles and ship's air.

Fuel Pump.—The fuel pump is practically similar to the M.A.N. design as built by the Nlseco Co., refined in detail. The plungers are vertical and carried on two crossheads driven by opposed cranks on a jack shaft which is driven by worm and gear from the crank shaft. The plungers are free on their stems and can be rotated with the fingers. The suction valves are measuring valves and are controlled by the fuel-control hand wheel and governor. There is also provided cut-out control from the starting levers and by air from the conning tower. The discharge valves are automatic checks, two for each pump. Small hand-priming plungers are provided for each cylinder on the fuel pump. They serve to prime the fuel lines before starting. A fuel reservoir is provided in the pump block, fitted with needle float valve and vent cock.

The fuel pump block is provided with eight pump barrels, the two middle ones being for oil (lubricating) pressure to the spray air regulator.

Governor.—The governor is of the simple inertia type and is driven off the forward end of the crank shaft. The inertia weights are small and the whole apparatus compact. It is

apparently an excess speed cut-off governor rather than a speed-control device.

Clutch.—The engine clutch is of the friction type and is air operated. The tail clutch is similar in design, but is operated by hand through worm and gear.

The outer casing or female member of the clutch is carried on the engine shaft, serving also as a flywheel. It is made in two parts, bolted together, and machined to cone surfaces inside, the cones being base to base.

The male part is carried on the motor shaft. The motor shaft carries a flange or spider on which are fixed four large pins parallel to the shaft axis and extending through the spider on either side. On these pins are carried two circular blocks, one forward and one abaft the spider. The blocks have a small axial freedom on the pins. The peripheral surfaces of the blocks are coned to conform to the inner surfaces of the fly wheel. Forcing the blocks apart engages the male and female conical surfaces. Drawing them together releases the surfaces and disengages the clutch. The cone blocks are operated from a sliding sleeve on the motor shaft by a series of links and toggles. The forward link of the system is a heavy steel spring. The conical surfaces are cut with oil grooves. The sleeve on the motor shaft is operated by means of a strong-back, fixed at one end and carried by the piston of an air cylinder at the other. Connection to the clutch sleeve is made by yoke and collar about midway of the strong-back.

Auxiliary operating gear is provided in a small hand air pump and air bottle.

The working parts of the clutch are all machine finished and finely fitted and its action is extremely smooth.

This apparatus was of special interest due to the compactness and small size for the power transmitted—1,200 horsepower.

Exhaust and Muffler.—The exhaust header is made up in three sections of steel with welded seams and flanges. It is

water-jacketed, with the usual jumpers around flanges. Exhaust from the cylinders is taken off by three elbows, each one serving two cylinders. The exhaust header is connected at its after end by a large water-jacketed pipe to the evaporator. From the after side of the evaporator another water-jacketed connection leads through the inboard exhaust valve to the elbow deck connection. At the after end of this elbow outboard is placed a large gate valve, and thence a long section of water-jacketed piping to the muffler. The muffler is a large cylindrical nonpressure expansion chamber. It is baffled by about six transverse plates with semicircular segments cut in their periphery. Segments in adjacent plates are staggered. The muffler is completely water-jacketed, but no water spray is put into the exhaust. The muffler is provided with a number of flapper vents, which are operated from inside the boat. A muffler drain is also provided with valve and reach rod into the motor room. It was noted that the entire exhaust system from the forward end of the exhaust header to the after end of the muffler is water-cooled. The inboard exhaust valves are large water-jacketed globe valves, with loose disks and auxiliary stems for grinding in without disassembly. These valves close against the sea pressure. The outboard exhaust valves are water-jacketed gate valves, situated outside the hull and operated from inboard by a system of reach rods. These are apparently meant to be quick closing.

Thrust Bearings.—Ball-bearing thrust bearings are installed on the line shafts. The installation is small and compact. The casing is made oil-tight, providing an oil bath for the bearing to run in.

Cam Shaft.—The cam shaft is a steel forging. It is carried in bearings whose pedestals are bolted to the intercylinder webs and flanges. The bearing structure is very rigid. The cam shaft rotates in the opposite direction from the crankshaft. It has an axial motion of about 1½ inches. Two cams for each valve are keyed to the cam shaft—one for ahead and

one for reverse. The axial motion of the cam shaft is obtained at the forward end by means of yoke and collar connection to the reversing gear. This motion is accommodated at the drive or after end by the shaft sliding in the drive gear wheel. Two diametrically opposite feathers of large size on the cam shaft, which fit in corresponding slots in the drive gear wheel, transmit the rotary motion.

Rocker Shaft.—The rocker shaft is a hollow steel forging which forms the pivots for all valve rocker arms. It is mounted in bearings carried by the same pedestals as the cam-shaft bearings. This shaft is made in sections for accessibility in lifting cylinder heads—six sections joined at the bearing supports.

The rocker shaft has a rotary motion of about 90 degrees for lifting the inlet and exhaust rockers off their respective cams when sliding the cam shaft on reversal. These valve rockers are pivoted on fixed eccentrics on the rocker shaft.

Carried on free eccentrics on the rocker shaft are the valve rockers for the spray and air-starting valves. These eccentrics are operated by the two starting levers, one controlling the three forward cylinders, the other the three after cylinders. Rotation of these eccentrics throws in the spray or starting rockers as desired. The middle position of these eccentrics leaves both valve rockers off their cams to permit translation of the cam shaft.

Air Starting Control.—For purposes of starting on air the engine is divided into two groups of three cylinders each with a control lever for each group. Both groups or the after group alone may be used for starting. The forward group only may be cut in one fuel while one group is on air. Actually operating the engines in the following practice was followed:

First step—both groups on air starting.

Second step—forward group on fuel.

Third step—after group on fuel.

These steps followed one another as fast as the starting levers could be operated, starting being accomplished in about four or five seconds.

The control lever for the after group: In the starting position—opens the relay of starting air valve, and also puts the eccentrics on the rocker shaft in the position where the starting-valve rockers are operative. It also operates the spray-spindle control to choke the spray air to the first group of cylinders. In the stop or mid-position—starting air valve is closed and the rocker shaft eccentrics are in mid-position with both spray and air-starting valve rockers clear of the cam shaft. In the running position—starting air valve is closed and the rocker-shaft eccentrics are in position to make the spray-valve rockers operative.

The control lever for the forward group: In the starting position, by means of a cam and small rocker shaft it holds down the fuel-measuring valves and cuts fuel off all cylinders. It also puts the rocker-shaft eccentrics in position where air-starting valve rockers are operative; in the stop or middle position it holds the rocker-shaft eccentrics in mid-position, so that both spray and starting-valve rockers are clear of the cam shaft; in the running position it sets the rocker-shaft eccentrics in position making the spray valve rockers operative, and releases the fuel-measuring valves.

The starting levers are interlocked with the reverse gear so that they can not be operated when the reverse gear is in any intermediate position. Conversely, the reversing gear can not be operated when the starting levers are in any but the stop position.

Reversing Gear.—The reversing gear is operated by means of a handwheel on a fore and aft horizontal shaft at the forward end of the engine. By means of a worm and gear, vertical shaft, bevel gears, and link motion, the rocker shaft is rotated to lift the inlet and exhaust rockers clear of their cams. Next by means of a yoke and collar connection the

cam shaft is moved axially about an inch and a half. The third and last step is to roll the rocker shaft back in position so that the inlet and exhaust valves are operative. The above steps are obtained by turning the handwheel eight complete turns. A sliding collar on the handwheel shaft is moved forward by means of a cam and spindle through the shaft while the gear is in any but final ahead or astern positions, and its function is by link connections to brake boxes to lift the cylinder relief valves during reversal.

Spray Spindle Control.—The lift of the spray-valve spindles is limited by a stop which is raised or lowered by means of a connection to the fuel-control handwheel. It operates to choke the spray air at low speeds and powers. Connection is also provided with the starting levers as explained above.

Fuel Control.—Control of the fuel-measuring valves is by means of interconnected links from the fuel handwheel and from the governor substantially the same as our practice. The control operates to hold open the fuel suction or measuring valves during any desired part of the pumping stroke of the plunger. A separate tappet plate is provided to hold the measure-valves completely open by link connection to the starting levers when they are in the "starting" or "stop" position. This tappet plate may also be operated by means of a small air plunger which is controlled by air connection from the conning tower.

Auxiliaries.—The attached oil pump is a multiple impeller pump which operates very quietly. The arrangement of holes in the idler gear and channels in its spindle appear to aid materially in the operation of the pump. It is believed they are arranged to relieve excess pressures where the gears mesh. The pump valves are arranged to accommodate running in either direction. This pump operates at the same revolutions per minute as the engines, being driven off the vertical shaft. The auxiliary oil pumps, two in number, are of similar type driven by separate motors.

The attached circulating water pumps are two-cylinder, single-acting, plunger pumps, driven by opposed cranks on a jack shaft driven by worm and gear from the crank shaft. On the four types of boats inspected these pumps were disconnected. The auxiliary-circulating water pumps, two in number, are of centrifugal type driven by separate motors on vertical shafts.

Fuel Compensation.—No fuel-supply pumps are installed; the fuel is displaced in the tanks by circulating water and forced to the gravity tanks. Compensation is automatic.

Coolers.—Both air and oil coolers are of the straight-tube type, installed with tubes vertical. There are duplicate oil coolers, and coolers for each stage of the engine air compressor. The air coolers on each engine are placed in a single shell, air entering at the top to the respective nests of tubes and passing out through separators at the bottom. Cooling water enters at the sides of coolers and passes across the tubes. It was not ascertained whether or not the water spaces were baffled.

Evaporator and Distiller.—An evaporator or heater is installed in the exhaust line abaft each engine. It is of the straight-tube type, tubes being placed horizontal and athwartships. Exhaust gases pass across the tubes. Evaporator feed is taken from the circulating water system. When not making water, the circulating water is passed through the evaporator coils with vapor pipe secured. The evaporator may be used as a heater for ship's heating and to proyide hot water for shower bath. One straight-tube distiller is provided for the two evaporators. It has a separate motor-driven pump for cooling water.

Strainers and Filters.—Strainers are of the basket type, similar to our practice. Filters are provided with muslin bags.

Instruments.—A striking feature of the M.A.N. engine is the permanent equipment of instruments provided. The engines are designed scientifically and installed to be operated

scientifically. Besides the usual equipment of gauges, thermometers are installed to give the temperature of practically every working and vital part of the engine.

A direct reading electric salinometer is installed to give instantaneous measurement of the chlorine content of evaporated water.

An electric apparatus for detecting salt water in the fuel supply is installed.

A differential pressure gauge is installed to indicate the oil pressure to and from the filter. This shows when the filter is restricted and needs cleaning.

Lubricating System.—Sump tanks are installed, located one under the center of each engine. Suction from the sump is taken to either or both attached or separately-driven oil pumps. From the pump the oil is delivered through a twin filter to the cooler. From the cooler the oil is divided between the lubricating and piston cooling systems. The lubricating oil is led to its pressure regulator and from the pressure regulator to a supply manifold pipe located on the inboard side of the crank case. Branches from the supply pipe are taken to each main bearing and to each cylinder. Oil from the main bearings is carried on to the crank-pin bearings and wrist pins from which overflow or leakage drips into the crank case. The lead to the main cylinder delivers oil which is picked up by the lower working piston ring which is fitted as a wiper.

The gear-set at the forward end of the engine is lubricated by connections to the discharge pipe from the pressure regulator.

The gear-set at the after end of the engine is lubricated by connections from the after end of the supply manifold pipe.

The rocker shaft is hollow and has a connection to the supply pipe which lubricates its bearings, the rocker bearings, roller bearings, rolls, and cam-shaft bearings. The cam shaft runs in an oil trough which is provided with drains to the return piston-cooling pipe.

The air-compressor cylinders are lubricated by means of a four-plunger lubricator driven from the countershaft.

Two drains are provided from the crank case to the sump tank, one located at each end of the crank case.

The lubricating oil found is very heavy.

Piston-Cooling System.—As noted above, the piston-cooling oil is branched off the oil discharge from the cooler, the bulk of the oil being taken by this branch. The lead is taken to a supply manifold pipe located along the inboard side of the crank case. Branches are taken off at each cylinder through the housing to the pistons and from the pistons return pipes lead to a sight drain box on the inboard side of the engine over the sump tank. The supply and return connections to the pistons are made up with knuckle joints, very similar to the installation on the *U. S. S. Fulton.*

Thermometers are fitted on the return oil leads to the sight drain box.

A by-pass between the oil-pump discharge and the crank-case drain passes through the piston-cooling oil regulator and serves to regulate the pressure in the cooling system by regulating the effective output of the pump.

Circulating Water System.—The circulating water system conforms to general practice.

Spray Air System.—The spray air system, air starting, and ship's air are interconnected and so used in German practice. The four-stage air compressors deliver air to the tanks at any required pressures up to about 2,500 pounds per square-inch. The spray flask also takes the compressor discharge pressure. From the spray flask a lead is taken to the spray air-pressure reducer and regulator and thence to the spray valves. Reliefs are fitted on the spray air lines. The spray air pressure is controlled statically by a spring-loaded reducer, in conjunction with which is an oil-pressure plunger which serves automatically to raise the spray air pressure as the engine speeds up. Pressure for the regulator plunger is obtained from a

two-plunger pump in the fuel-pump block. Lubricating oil is used for a pressure medium.

Fuel System.—Fuel is supplied to the gravity tanks under the pressure of the circulating water system, circulating water being used to displace the fuel oil and thereby also compensate for the weight of fuel used.

The fuel found is very light, about the consistency of kerosene, and smells like kerosene.

Indicator Gear.—Indicator gear is provided for taking power cards. The indicator cock connection is attached to the inboard side of each cylinder head. The stroke motion is obtained from the piston-cooling return link motion by means of a small spindle which passes up through an oil-tight sleeve in the housing.

Air-compressor cylinders are also provided for taking cards.

Workmanship.—The workmanship on the engines is uniformly excellent. The material itself is uniformly fine in structure and shows utmost skill in manufacture. All working parts are machine finished and accurately fitted. The engines are apparently designed in every detail to produce a finished product, each detail contributing to the final result. No makeshift jobs or evidences of afterthought were visible, as is often the case in less scientific machines.

Practically all nuts on the engine are wrench-tight on their bolts or studs the full length of the thread, and all are secured by means of pins, wires, or locking washers. Handwheels on high-pressure air lines are large and fitted with a quarter to a half turn of lost motion on their stems to give a slight striking force for opening or closing. Pipe flanges which require frequent disassembly are made up with portable fastenings. The flanges are slotted in place of drilled bolt holes and the bolts are hinged under the lower flange.

Air bottles are of steel, two spray flasks and two starting bottles being supplied for two 1,200-horsepower engines. The combined capacity of the starting bottles is about 35 cubic feet.

Due to rigid construction and fine balance, the engines operate with extremely little vibration and are comparatively quiet. The engine-room air is free of smoke and gases and the exhaust is perfectly clear and well muffled.

CURRENT-CARRYING CAPACITY OF ELECTRIC CABLES, U. S. NAVY.

Calculations of Wire and Fuse Sizes.

By Lieut.-Comdr. Alexander M. Charlton, U. S. Navy, Member.

When electricity was first introduced in the service the current-carrying capacity of electric cable was specified as not over 1,000 ampères per square inch cross section of conductor. This was equivalent to 1,273 circular mils per ampère of current carried.

Later the specifications were changed to call for 1,000 circular mils per ampère for continuous loading and 500 circular mils per ampère for intermittent loading.

Both of the above specifications were arbitrary assumptions considered to give a factor of safety which would eliminate the possibility of overheating the cables. No consideration was given to initial temperature or to temperature rises. As these specifications were coupled with a percentage of voltage drop which in most cases was the limiting factor, no serious difficulty arose, as the minimum drop allowed usually required a larger cable than was necessary to carry the current. These arbitrary specifications based on experience with comparatively small cables had no scientific basis and as the size of cables increased aboard ship occasional cases of overheated cables were reported.

As will be shown later, we were underloading our cables below the 200,000 circular mil size and overloading them above this size for temperature conditions now considered as safe.

In 1914 the Bureau of Steam Engineering requested the Navy Yard, New York, to conduct tests to determine the current-carrying capacity of Navy standard cables.

In test Number 175, dated December 16, 1915, the Navy Yard reported tests on a number of leaded and armored cables taken from stock, the following sizes being used:

Single Conductor.	*Twin Conductor.*
9,030 circular mils.	4,494 circular mils.
75,850 circular mils.	9,030 circular mils.
98,820 circular mils.	14,350 circular mils.
198,860 circular mils.	22,820 circular mils.
521,970 circular mils.	49,020 circular mils.
	59,940 circular mils.

The cables tested were suspended in a horizontal position four inches from the floor, exposed on all sides to the air. The cables, which were from twenty to thirty feet long, with copper terminals soldered to the ends, were tested one at a time, the ends being connected directly to a motor generator set.

The tests were made for a temperature rise of 30 degrees C. commencing with a room temperature of 25 degrees C. (77 degrees F. to 131 degrees F.). The following results were obtained:

Size of cables, c. m. s.	Ampères rating (Single Conductor)	Ampères rating (Twin Conductor)
4,494	.....	26
9,030	44.5	40
14,350	.....	57
22,820	.....	74
49,020	.....	111
59,940	.....	128
75,850	178.0	...
98,820	225.0	...
198,860	325.0	...
521,970	656.0	...

Plotting these results we get a curve which varies considerably from the Navy specification of 1,000 circular mils per ampère. As will be seen later, the initial temperature used in this test was considered by the Bureau too low, and the allowed temperature rise too great for conditions aboard ship. Also, no grouping and location factor was allowed for. The values obtained by New York, however, agree closely with those in the standard table when reduced to the same limits.

The current which a cable will carry without reaching a temperature dangerous to the copper or insulation depends upon the establishment of a balance between the heat generated by the flow of current through the conductor and the heat conveyed from the conductor to the insulation and thence radiated to the atmosphere. The area over which the dissipation of heat must take place depends upon the circumference of the cable, that is, it is a function of the diameter. The heat generated varies as the square of the current, and inversely as the square of the diameter of the cable.

Resistance $= \rho \frac{1}{a}$ and $a = \frac{\pi d^2}{4}$. We would expect then that the current allowed for a certain temperature rise would vary as the diameter to the 3/2 power. In consequence the amperage per circular mil of cross-section must be less for the large size conductors than for the small ones in order to maintain the same temperature rise.

The formula worked out by the New York Navy Yard for this particular test is

$$I = .015 \sqrt{\frac{d^3}{x}} \text{ where}$$

d = diameter of copper in mils

x = thickness of insulation including tape and braid

.015 = constant depending on temperature conditions, conductivity of insulation, etc.

Further tests by the Navy Yard, New York, gave additional formulæ arranged in this form:

$I = .01483 \sqrt{\frac{d^3}{x}}$ 30° C. rise. Final temperature, 63° C.

$I = .0121 \sqrt{\frac{d^3}{x}}$ 20° C. rise. Final temperature, 65° C.

$I = .01373 \sqrt{\frac{d^3}{x}}$ 26 2/3° C. rise. Final temperature, 70° C.

These formulæ were derived as follows:

Watts loss per foot length of cable $= \frac{I^2 r}{d^2}$

r = resistance in ohms per mil foot at final temperature.

d = diameter of copper in mils (for standard cable use diameter of solid wire of equivalent area).

I = current in ampères.

Watts radiated per foot $= \frac{12\,\Pi d}{1000} \times \frac{Tk}{x}$

T = temperature rise in degrees C.

x = thickness of insulation including tape and braid.

k = constant depending on the thermal conductivity of the insulation and the radiating capacity of the outer surface of the insulation or armor.

The average value of k from test Number 175 is .00243.

Equating these two

$$\frac{I^2 r}{d^2} = \frac{12\,\Pi d}{1000} \frac{Tk}{x}$$

$$I = \sqrt{\frac{12\,\Pi T k}{1000\ r} \frac{d^3}{x}}$$

It was realized that the above formulæ were not in strict accordance with theory, but they were more convenient than the purely theoretical formulæ and were considered to give

results which were conservative and within the limits to which the radiation constants could be determined.

At the request of the Navy Yard, New York, Professor Slichter, of Columbia University, checked up the figures of the Navy Yard by comparison with data prepared by Powell and by Langmuir and Dushman.

Professor Slichter stated that the figures proposed by the Navy Yard "— are very reasonable and in accordance with results obtained by other investigators."

At the outbreak of war, the Standards Committee of the American Institute of Electrical Engineers offered its services to the Government. The Bureau of Steam Engineering requested consideration of A.C. high-tension cables proposed for electric propulsion, and subsequently suggested to the committee that the problems of current-carrying capacity of low-tension cables, limiting temperatures and insulation characteristics, could be advantageously considered by the Standards Committee and results obtained which could be used as a standard for the Naval service.

The sub-committee on wires and cables of the Standards Committee undertook the consideration of these problems. The sub-committee was composed of representatives of the Bureau of Steam Engineering and the Navy Yard, New York, and representatives of the following companies, among others:

The Standard Underground Cable Co.
The Habirshaw Electric Cable Co.
The Safety Insulated Wire and Cable Co.
The Simplex Wire and Cable Co.
The General Electric Co.
The Westinghouse Electric and Manufacturing Co.
The New York Edison Co.
The Public Service Corporation of New Jersey.

The report of the committee on D.C. cables was prepared by Mr. Philip Torchio, Chief Electrical Engineer of the New York Edison Co.

The Bureau requested ratings based on the current-carrying capacities of cables under three conditions:

I. Continuous current rating.

II. Thirty-minute rating—starting from the end of the continuous rating run.

III. Thirty-minute rating—starting from room temperature.

In order to simulate conditions aboard ship, all ratings were requested to be based on an ambient temperature of 45 degrees C. (113 degrees F.). This meets fire and engine room requirements in a measure and also takes care of tropical temperatures.

The allowable temperature rise for the various cables under different ratings is as follows:

	Temperature Rise.	
	Rubber Insulation.	Varnished Cambric Insulation.
I. Continuous rating.....	15° C.	30° C.
II. Thirty-minute rating after continuous run..	25° C.	30° C.
III. Thirty-minute rating starting from room temperature	25° C.	30° C.

Due to the high initial temperature selected as a basis, the temperature rises were necessarily kept low in order to avoid reaching temperatures where the insulation would begin to be affected. The final temperature for varnished cambric insulated and for reinforced rubber insulated cable is 75 degrees C. (167 degrees F.). As these types of cable are specified for use in machinery and boiler spaces, the probable maximum temperatures are provided for.

The values for the various ratings were obtained from the following formulæ:

I. CONTINUOUS RATING.

The energy lost in the cable $= I^2r$ watts. This multiplied by $(h + J)$ equals the temperature rise or $T = I^2r\ (h + J)$

and $$I = \sqrt{\frac{T}{r\ (h + J)}}.$$

To allow for the lack of heat dissipation due to the proximity of other conductors, location of conductors close to decks, etc., it was decided to limit the current to 80 per cent of the maximum current-carrying capacity, and

$$I_e = .8 \sqrt{\frac{T}{r\ (h + J)}}.$$

Where I = maximum current

I_e = maximum allowable current

T = permissible temperature rise (15 degrees C. for rubber, 30 degrees C. for varnished cambric)

r = resistance in ohms per inch of conductor

h = heat resistance from copper to armor in degrees C. rise per watt per inch of conductor

J = heat resistance from armor to room temperature in degrees C. rise per watt per inch of conductor

.8 = grouping factor considering that four cables may be racked together in a location with poor ventilation such as under deck or adjacent to bulkheads.

The factor 'h' above is expressed by the general equation

$$h = \frac{K}{2\Pi} \log_e \frac{b}{a} + \frac{K^1}{2\Pi} \log_e \frac{c}{b} \text{ etc.}$$

Where a, b, c, d are the diameters of the various belts of insulation and K, K^1, etc., are constants representing the heat resistivity of the various materials in degrees C. rise per watt per inch cube of the materials.

This formula may be derived as follows:

$$\text{Resistance} = \text{specific resistance} \times \frac{\text{length}}{\text{area}}$$

For a unit of cable the length of path of heat flow from the inside out $= dx$. The area of this path $= 2\Pi x$

$$h = \int_{\frac{a}{2}}^{\frac{b}{2}} \frac{K dx}{2\Pi x}$$

When $b = a$, resistance $= o$

Therefore $C = o$ and

$$h = \frac{K}{2\Pi} \log_e \frac{b}{a} \text{ etc.}$$

II. THIRTY-MINUTE RATING—STARTING FROM THE END OF THE CONTINUOUS RUN.

The energy lost in the conductor in watts in any given time is equal to the watts radiated plus the watts absorbed in the cable due to its thermal capacity.

$$\text{Therefore } \left(\frac{I}{.8}\right)^2 r = \frac{T}{(h + J)} + \frac{w d'T}{dt}$$

Where w = the heat input in watt seconds required to raise copper 1 degree C. per second and the insulating materials, lead and armor in proportion to their temperature rise for 1 degree C. increase in copper temperature

t = time in seconds.

$$\frac{d'T}{\left(\frac{I}{.8}\right)^2 r - \frac{T}{(h+J)}} = \frac{dt}{w}$$

At the beginning of the thirty-minute run

$$t = o \text{ and } T = \left(\frac{I_o}{.8}\right)^2 r\,(h + J) \text{ and } I = I_o$$

$$\text{Let } P_o = \left(\frac{I_o}{.8}\right)^2 r. \quad \text{Then } T = P_o\,(h + J).$$

At the end of the thirty-minute run $t = 1{,}800$ and $T = T_r$ = maximum allowable temperature.

Integrating, with these limits of integration

$$T_r = (h + J)\left(\frac{I}{.8}\right)^2 r\left[1 - \cosh\frac{1800}{w(h+J)} + \sinh\frac{1800}{w(h+J)}\right]$$

$$+ P_o\,(h + J)\left[\cosh\frac{1800}{w(h+J)} - \sinh\frac{1800}{w(h+J)}\right]$$

III. THIRTY-MINUTE RATING STARTING FROM ROOM TEMPERATURE.

The same differential as in II expresses the relation existing, but in this case the limits are $T = T_r$ and $T = o$ and $t = 1{,}800$ and $t = o$.

$$\int_o^{T_r} \frac{dT}{\frac{I^2}{.8}r - \frac{T}{(h+J)}} = \int_o^{1800} \frac{dt}{w}$$

$$T_r = (h + J)\left(\frac{I}{.8}\right)^2 r\left[1 - \cosh\frac{1800}{w(h+J)} + \sinh\frac{1800}{w(h+J)}\right]$$

In arriving at these formulæ two assumptions are made:

(*a*) That the resistance r is a constant—this is not strictly correct, as r is a function of T due to the temperature coefficient of the conductor.

(*b*) That the heat resistance J from armor to air is a constant. This is not strictly correct.

These discrepancies in a measure offset each other and are not of great importance with the low temperature rises under consideration. Of the above formulæ, I and III only are used in calculating wire sizes for use on Naval vessels, I for cables

carrying current continuously, and III for cables which will not carry maximum current for periods longer than thirty minutes at a time.

The following values for the constants used in making up the tables were determined by experimental work by the Navy Yard, New York, and by the New York Edison Co.:

$H = 116.5 \quad H = \Pi d J$ where d = diameter of cable overall.

$$J = \frac{H}{\Pi d}$$

K rubber = 285
K varnished cambric = 295
K reinforced rubber = 220
K lead to steel = 91
K lead = 1.23

From Bureau of Standards Circular 31, October 1, 1914, the resistance of copper = 10.371 ohms per mil foot at 20 degrees C., and 100 per cent conductivity. r = 1.0177 ohms/mil inch at 60 degrees C. and 98 per cent conductivity, and r = 1.0686 ohms/mil inch at 75 degrees C. and 98 per cent conductivity.

For determining values of w, the following densities were used:

Copper	.1142 grams per 1,000 circular mil inch.
Rubber	28.7 grams per cubic inch.
Lead	186.0 grams per cubic inch.
Steel	126.0 grams per cubic inch.
Varnished cambric	19.0 grams per cubic inch.
Reinforced rubber	21.4 grams per cubic inch.

The specific heats used were

Copper	= .393	Watt-seconds per gram per degrees C. rise.
Rubber	= .782	
Varnished cambric	= 1.672	
Lead	= .13	
Steel braid	= .494	

The value of w in formula III is made up as follows:

w_1 for copper = weight per inch $\times$ specific heat $\times$ 1
w_2 for insulation = weight per inch $\times$ specific heat $\times$ .85
w_3 for lead = weight per inch $\times$ specific heat $\times$ .70
w_4 for steel braid = weight per inch $\times$ specific heat $\times$ .65
$w = w_1 + w_2 + w_3 + w_4$

The factors 1, .85, .70 and .65 used in the above expressions indicate the temperature rise of the several materials per degrees C. rise in the copper.

The examples below are worked out to show the method of constructing the table.

414,020 CIRCULAR MIL RUBBER COVERED LEADED AND ARMORED CABLE CONTINUOUS CURRENT RATING.

r (ohms per inch at 60 degrees C.) $= \frac{1.0177}{414{,}020} = 2.458 \times 10^{-6}$

Conductor diameter $(2a)$ ..... = .742 inch
Diameter over rubber $(2b)$ ..... = 1.066
$\frac{2b}{2a}$ ..... = 1.437
$\frac{K}{2\Pi} \log_e \frac{2b}{2a} = \frac{285}{2\Pi} \log_e 1.437$ ..... = 16.43
Diameter over lead $(2c)$ ..... = 1.226
$\frac{2c}{2b}$ ..... = 1.15
$\frac{K_1}{2\Pi} \log_e \frac{2c}{2b} = \frac{1.23}{2\Pi} \log_e 1.15$ ..... = .027
Diameter overall $(2d)$ ..... = 1.396
$\frac{2d}{2c}$ ..... = 1.138
$\frac{K_2}{2\Pi} \log_e \frac{2d}{2c} = \frac{91}{2\Pi} \log_e 1.138$ ..... = 1.87
$h = 16.43 + .027 + 1.87$ ..... = 18.33

$$J = \frac{H}{\Pi d} = \frac{116.5}{\Pi \times 1.396} \ldots\ldots\ldots\ldots\ldots\ldots = 26.60$$

$$(h + J) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots = 44.93$$

$$I = .8\sqrt{\frac{T}{r(h+J)}} = .8\sqrt{\frac{15}{2.458 \times 10^{-6} \times 44.93}} = 294.5 \text{ amps.}$$

THIRTY-MINUTE RATING STARTING FROM ROOM TEMPERATURE.

$$I = .8\sqrt{\frac{T_r}{r(h+J)\left[1 - \cosh\frac{1800}{w(h+J)} + \sinh\frac{1800}{w(h+J)}\right]}}$$

$$r = 2.54 \times 10^{-6}$$

$h + J = 44.93$ from above

$$w_1 = .1142 \times 414.020 \times .393 \ldots\ldots\ldots\ldots = 18.55$$

$$w_2 = 28.7 \times \frac{\Pi}{4}(1.066^2 - .742^2) \times .782 \times .85 = 8.76$$

$$w_3 = 186 \times \frac{\Pi}{4}(1.226^2 - 1.066^2) \times .13 \times .70 = 4.87$$

$$w_4 = 126 \times \frac{\Pi}{4}(1.396^2 - 1.226^2) \times .494 \times .65 = 14.20$$

$$w = \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 46.38$$

$$\frac{1{,}800}{w(h+J)} = \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots .867$$

$$I = .8\sqrt{\frac{25}{2.54 \times 10^{-6} \times 44.93(1 - 1.40 + .98)}} = 493 \text{ amps.}$$

30,780 CIRCULAR MIL CABLE LEADED AND ARMORED VARNISHED CAMBRIC INSULATION.

CONTINUOUS CURRENT RATING.

r (ohms per inch at 75 degrees C.) $\ldots\ldots = 34.71 \times 10^{-8}$

Conductor diameter $(2a) \ldots\ldots\ldots\ldots\ldots = .201$ inch

Diameter over tape $(2b) \ldots\ldots\ldots\ldots\ldots = .383$

$$\frac{2b}{2a} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots = 1.91$$

$$\frac{K}{2\Pi}\log_e\frac{2b}{2a} = \frac{295}{2\Pi}\log_e 1.91 \ldots\ldots\ldots\ldots = 30.8$$

Diameter over lead sheath $(2c) \ldots\ldots\ldots\ldots = .453$ inch

$$\frac{2c}{2b} \ldots\ldots = 1.185$$

$$\frac{K}{2\Pi} \log_e \frac{2c}{2b} = \frac{1.23}{2\Pi} \log_e 1.185 \ldots\ldots = .033$$

Diameter overall $(2d) \ldots\ldots = .623$

$$\frac{2d}{2c} \ldots\ldots = 1.378$$

$$\frac{K}{2\Pi} \log_e \frac{2d}{2c} = \frac{91}{2\Pi} \log_e 1.378 \ldots\ldots = 4.63$$

$$h = 30.8 + .033 + 4.63 \ldots\ldots = 35.46$$

$$J = \frac{H}{\Pi d} = \frac{116.5}{\Pi \times .623} \ldots\ldots = 59.50$$

$$h + J \ldots\ldots = 94.96$$

$$I = .8\sqrt{\frac{30}{34.71 \times 10^{-6} \times 94.96}} = 76.5 \text{ ampères.}$$

THIRTY-MINUTE RATING.

$$r = 34.71 \times 10^6$$

$$h + J = 94.96$$

$$w_1 = .1142 \times 30.780 \times .393 \ldots\ldots = 1.382$$

$$w_2 = 19 \times \frac{\Pi}{4} (.383^2 - .201^2) \times 1.672 \times .85 = 2.250$$

$$w_3 = 186 \times \frac{\Pi}{4} (.453^2 - .383^2) \times .13 \times .70 = .785$$

$$w_4 = 126 \times \frac{\Pi}{4} (.623^2 - .453^2) \times .494 \times .65 = 5.810$$

$$w = w_1 + w_2 + w_3 + w_4 \ldots\ldots = 10.227$$

$$I = .8\sqrt{\frac{30}{34.71 \times 10^{-6} \times 94.96 \left[1 - \cosh \frac{1800}{10.23 \times 94.96} + \sinh \frac{1800}{10.23 \times 94.96}\right]}} = 83.2 \text{ ampères.}$$

The values worked out in the examples above agree very closely with those given in the table. In making up the tables the various currents were worked out and the results plotted on logarithmic paper, using as coordinates amperes versus size of conductor in circular mils. The values for a particular type of cable will lie practically on a straight line. This is quite true for the continuous current values where the relation is practically a simple variation of current with circular mil area.

For the intermittent ratings the relation between circular mil area and current-carrying capacity is not quite a straight line, due to the greater relative thermal capacity of the copper, armor and insulating materials of the larger-sized cable. This would tend to make the curve depart upward from the straight line.

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

RUBBER INSULATION CABLES - LEADED AND ARMORED

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	18	15	25	20
9,000	9,050	26	22	36	31
11,000	11,340	31	25	41	35
14,000	14,350	35	30	47	40
18,000	18,060	41	35	55	47
23,000	22,820	47	40	64	54
30,000	30,780	56	48	77	65
40,000	38,950	66	56	88	75
50,000	49,020	75	64	102	87
60,000	59,940	85	72	117	99
75,000	75,850	99		141	
100,000	98,820	118		172	
125,000	125,050	135		205	
150,000	157,380	150		240	
200,000	198,860	184		283	
250,000	250,710	214		347	
300,000	296,660	238		382	
375,000	374,010	280		455	
400,000	414,080	295		495	
500,000	521,970	344		585	
650,000	657,860	402		715	
800,000	829,310	475		835	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

RUBBER INSULATION CABLES - ARMORED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	17	14	19	16
9,000	9,030	24	20	30	25
11,000	11,340	28	24	34	29
14,000	14,350	32	27	39	33
18,000	18,060	38	32	46	39
23,000	22,820	45	37	54	46
30,000	30,780	52	44	65	55
40,000	38,950	61	52	74	63
50,000	49,020	69	59	86	73
60,000	59,940	78	66	99	84
75,000	75,850	92		119	
100,000	98,820	110		148	
125,000	125,050	127		175	
150,000	157,380	149		206	
200,000	198,860	175		246	
250,000	250,710	204		303	
300,000	296,560	226		356	
375,000	374,010	255		403	
400,000	414,020	280		437	
500,000	521,970	327		520	
650,000	657,860	383		662	
800,000	829,510	451		785	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

RUBBER INSULATION CABLES - PLAIN BRAIDED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	16	14	18	15
9,000	9,030	22	19	28	24
11,000	11,340	27	23	32	27
14,000	14,350	30	26	37	31
18,000	18,060	36	31	43	37
23,000	22,820	44	37	50	43
30,000	30,780	49	42	61	52
40,000	38,950	58	49	70	60
50,000	49,020	66	56	81	69
60,000	59,940	70	60	93	79
75,000	75,850	87		113	
100,000	98,820	106		138	
125,000	125,050	120		165	
150,000	157,380	142		195	
200,000	198,860	167		232	
250,000	250,710	195		286	
300,000	296,560	218		317	
375,000	374,010	256		382	
400,000	414,020	270		420	
500,000	521,970	316		503	
650,000	657,860	378		634	
800,000	829,510	441		767	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC CABLES - LEADED AND ARMORED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	24	20	25	21
9,000	9,030	36	31	38	32
11,000	11,340	42	36	45	38
14,000	14,350	48	41	52	44
18,000	18,060	56	48	61	52
23,000	22,820	64	54	70	60
30,000	30,780	77	65	86	73
40,000	38,950	90	77	100	85
50,000	49,080	104	88	117	99
60,000	59,940	117	99	134	114
75,000	75,850	136		158	
100,000	98,820	162		189	
125,000	125,050	187		223	
150,000	157,380	218		266	
200,000	196,860	252		316	
250,000	250,710	294		380	
300,000	296,660	326		432	
375,000	374,010	382		515	
400,000	414,020	409		560	
500,000	521,970	474		668	
650,000	657,860	555		790	
800,000	829,310	660		950	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC CABLES - ARMORED.

Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	25	20	25	20
9,000	9,030	33	28	34	29
11,000	11,340	39	33	40	34
14,000	14,350	45	38	46	39
18,000	18,060	52	44	54	46
23,000	22,820	60	51	62	53
30,000	30,780	72	61	77	65
40,000	38,950	84	71	89	76
50,000	49,080	97	82	104	88
60,000	59,940	109	93	120	102
75,000	75,850	127		142	
100,000	98,820	151		170	
125,000	125,050	174		208	
150,000	157,380	204		242	
200,000	196,860	237		291	
250,000	250,710	278		350	
300,000	296,660	308		398	
375,000	374,010	361		474	
400,000	414,020	388		515	
500,000	521,970	450		615	
650,000	657,860	530		736	
800,000	829,310	625		905	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC CABLES - PLAIN BRAIDED.

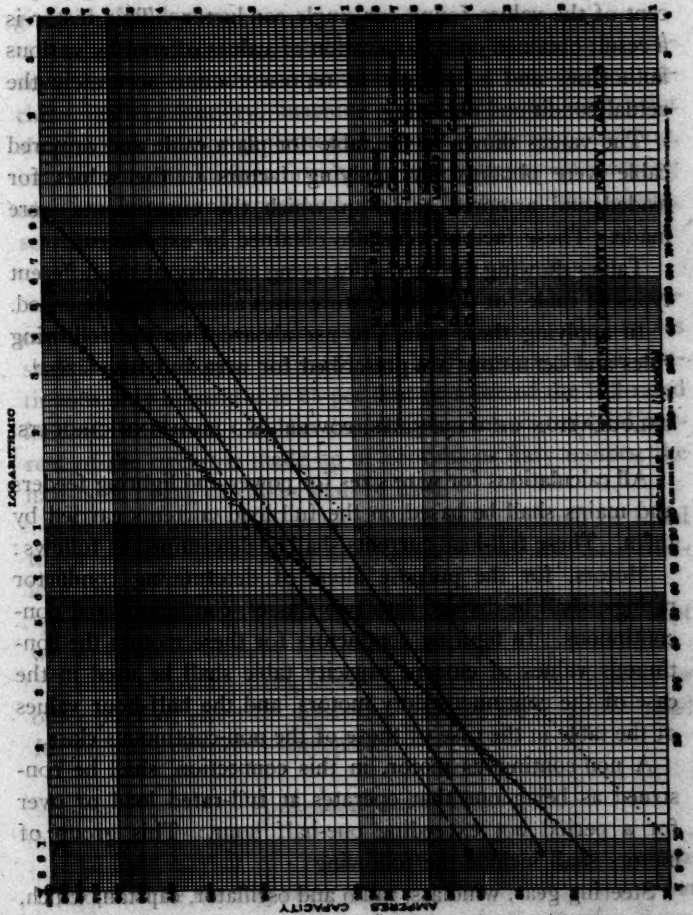
Circular Mils		Continuous Load		Intermittent Load	
Approximate	Actual	Single	Duplex	Single	Duplex
4,000	4,494	21	18	21	18
9,000	9,030	31	26	31	26
11,000	11,340	36	31	36	31
14,000	14,350	42	36	42	36
18,000	18,060	49	42	49	42
23,000	22,820	56	48	57	49
30,000	30,780	67	57	70	60
40,000	38,950	79	67	81	69
50,000	49,020	91	77	95	81
60,000	59,940	96	82	110	94
75,000	75,850	113		130	
100,000	98,820	144		158	
125,000	125,050	166		188	
150,000	157,380	196		228	
200,000	198,860	230		274	
250,000	250,710	268		330	
300,000	296,660	298		378	
375,000	374,010	350		450	
400,000	414,020	375		490	
500,000	521,970	435		588	
650,000	657,860	513		705	
800,000	829,510	605		865	

TABLES SHOWING MAXIMUM ALLOWABLE LOAD IN AMPERES FOR CONTINUOUS AND INTERMITTENT LOADS ON NAVY STANDARD CABLES.

VARNISHED CAMBRIC WITH REINFORCED RUBBER CABLES.

Circular Mils		Armored		Without Armor	
Approximate	Actual	Continuous Single	Intermittent Single	Continuous Single	Intermittent Single
250,000	250,710	280	376	272	357
300,000	296,660	312	429	302	409
375,000	374,010	362	512	352	490
400,000	414,020	387	552	376	528
500,000	521,970	449	660	436	634
650,000	657,860	525	790	510	757
800,000	829,510	619	940	610	902

These relations are shown on the curves attached.



The table in its final form was picked off from the faired curve, which explains the small discrepancies noted in the examples worked out.

The table for twin conductor was made up by taking 85 per cent of the values found for single conductor. This factor is a result obtained on consideration of values offered by various authorities and is thought to represent quite accurately the actual conditions obtaining.

The values shown in the table for plain cable and armored cable were obtained by applying factors to the values for leaded and armored cable on which the experiments were made. These factors were also obtained by experiment.

Tables showing the current-carrying capacity of the different types of cable for the temperature rises allowed are appended.

In applying these values to use aboard ship the following rules and definitions are prescribed for wire and fuse sizes:

CALCULATION OF WIRE SIZES, POWER AND LIGHTING CIRCUITS.

All calculations for wire sizes for power and lighting feeders and mains shall be based on the full load currents carried by them. These full-load currents shall be determined as follows:

Motors, for the purpose alone of determining conductor ratings, shall be divided into two classes, continuous and non-continuous. In figuring conductors for these motors the continuous values of current capacity table shall be used in the case of the continuous duty motors, and the half-hour values of the table in the case of those of the non-continuous class.

A non-continuous motor, in this connection, shall be construed as being one that operates at full-rated load or over for a period not exceeding one-half hour. This group of motors shall include the following:

Steering gear, windlass, radio and oscillator, capstan, winch, turret machinery, boat crane, turbine and shaft turning, torpedo hoist, and sounding machine motors.

Continuous motors shall include in general those not plainly of the non-continuous class and, specifically, those which operate continuously at times at full-rated load for periods of over one-half hour. A list of such motors and apparatus is given for guidance:

Ventilation, blower, portable fan, ammunition hoist, conveyor, pump, engine-room auxiliary, air compressor, ice machine, galley, laundry, machine shop, workshop and print shop motors; motor-generators for battery charging, welding, interior communication, fire control and submersible pumping apparatus, etc.; and apparatus such as heaters, stoves, ovens, searchlights.

Auxiliary power shall generally be rated continuous.

In power feeders or mains when both continuous and non-continuous duty motors are used, the continuous rated cables shall be used in case the rated full-load capacity of the continuous duty apparatus is 25 per cent or more of the full-rated load of the circuit. If less than 25 per cent of full-load current of the circuit is made up of continuous duty motors, the half-hour rating of cables shall be used.

The rated ampère capacity of cable selected shall be equal to, or in case of excessive drop, greater than the rated full-load current of the circuit.

In selection of all cables, the drop in voltage shall be kept within limits of existing specifications.

In the following paragraphs the rated load of a motor or other piece of electrical apparatus shall be the ampères called for on manufacturer's nameplate.

FULL-LOAD CURRENT.

Full-load current of circuit between generator and switchboard shall be the full-rated load of generator plus 33 1/3 per cent, and between generator board and distribution board shall be the full-load rating of the several ship's service gen-

erators in one engine room, omitting exciter, if any, plus 33 1/3 per cent.

The combined *full-load current* of power and lighting-bus tie feeders between distribution boards in cases where lighting and power cables constitute independent circuits shall be equal to rated capacity of ship's service generators in one dynamo or engine room.

The *full-load current* of bus tie cables between distribution boards when these cables carry jointly the lighting and power loads shall be that equivalent to one-half of the overload capacity (133 per cent) of ship's service generators in one engine or dynamo room.

Bus tie feeders between generator boards and distribution boards and between forward and after distribution boards, and also generator cables, shall be figures so as to observe the following conditions:

(*a*) Drop in voltage on power circuits from generator board to motor feeding from its own distribution board shall not exceed 5 per cent, and in case of motor feeding from opposite generator board, 6 per cent.

(*b*) Wiring for lighting systems will be designed on the basis of a maximum allowable drop of 2½ per cent, calculated from adjacent generator switchboard, and 3 per cent from the distant generator switchboard, based on calculated full-load currents hereinafter defined.

(*c*) Maximum allowable drop, generator to distribution board, 1 per cent.

(*d*) Allowable drop, generator board to adjacent distribution board, one-half of 1 per cent.

(*e*) The allowable drop in voltage over such feeders between distribution boards as shall be utilized for lighting service shall not exceed one-half of 1 per cent, based on a load equivalent to 1/3 of the sum of the *full-load currents* of all lighting feeders of forward or after lighting systems. This shall be considered the actual lighting load in the following paragraph:

(*f*) The allowable drop in voltage due to power load over the remainder of these feeders between distribution boards shall not exceed 1 per cent, a load current being assumed equivalent to the total generator load described above less the actual lighting load described in (*e*).

(*g*) In cases where one set of bus tie feeders between distribution boards carries jointly lighting and power load, the drop in voltage, under *full-load current* condition specified for this arrangement in paragraph above, shall not exceed 1 per cent.

(*h*) The total cross-section of all conductors between generator board and distribution board shall be at least equal to the total cross-section of cables between generator and generator board.

(*i*) In cases where the distribution panels and the generator panels are integral parts of the same switchboards the drop in voltage between generators and generator panels shall not exceed one-half of 1 per cent.

The *full-load current* of a feeder or main running to but one motor shall be the rated load of the motor.

The *full-load current* of a feeder or main running to a heating device or heating devices shall be the full-load current of the heating device or the sum of the full-load currents of the several heating devices.

The *full-load current* of a main running to several motors and heating devices shall be the sum of the full-load rating of the motors and heating devices plus the sum of the current capacities of the additional outlets on the main, if any.

The *full-load current* of a feeder from distribution board to an auxiliary power panel shall be the sum of the full-load currents of the mains running from the panel plus the sum of the current capacities of spare outlets or switches on the panel.

The *full-load current* of a feeder running direct to junction boxes supplying several motors and heating devices with no

auxiliary panel shall be the sum of the full-load currents of the motors and heating devices.

The *full-load current* of a lighting feeder running direct to a lighting panel or to feeder junction boxes will be equal to the sum of the full-load currents of the lighting mains feeding from the panel or junction boxes.

The *full-load current* of a lighting main running from a lighting panel or feeder junction box will be equal to the sum of the full-load currents of all sub-mains and branches feeding from it.

The *full-load current* of a lighting sub-main will be equal to the sum of the full-load currents of all branches feeding from the distribution box or junction box.

Lighting feeders to lighting panels or to feeder junction boxes will in no case be less than 9,030 c.m. twin conductor cable.

Lighting cables will in no case be less than 4,494 c.m. twin conductor, except in special cases such as signal, running and anchor light systems. The lighting sub-mains to distribution boxes should be of such cross-section that the drops in them will not cause undue increases in sizes of feeders and mains.

FUSE SIZES.

The circuits included in the accompanying rules will comprise feeders, mains and sub-mains. The following definitions are given as applying in this instance:

(*a*) A feeder shall be considered as a circuit fed directly from main power or lighting distribution board.

(*b*) A main shall be considered as a circuit receiving its energy directly from a feeder.

(*c*) A sub-main shall be considered as a circuit feeding directly from a main and supplying one or more distribution boxes.

The rated capacity of fuse for a power feeder from distribution board shall be equal to the full-load current of the

circuit plus 50 per cent of the current of largest motor, in no case, however, to be less than 25 per cent in excess of total rated ampères of connected load.

The rated capacity of fuse for lighting feeders from lighting distribution board shall be equal to the full-load current of the circuit plus 25 per cent.

The rated capacity of fuse on power mains and sub-mains shall be equal to the full-load current of such mains or sub-mains plus 50 per cent of the current of largest motor, in no case, however, to be less than 25 per cent in excess of total rated ampères of connected load.

The rated capacity of fuse on lighting mains or sub-mains shall be equal to the full-load current of such mains or sub-mains.

The rated capacity of power mains or sub-mains carrying heaters alone shall be equal to the combined full-load current of the connected heaters.

With the table of current-carrying capacities given and the rules for applying these tables, it is believed that no copper will be wasted in laying out the various systems aboard ship while there will be sufficient copper provided to carry the desired currents without overheating.

U. S. S. *EAGLE* BOATS.

DESCRIPTION.

By Commander Carlos Bean, U. S. Navy, Member.

(Plates Nos. 1 to 6 show construction of hull and launching devices.)

In February, 1918, the Ford Motor Company, of Detroit, Michigan, contracted to build one hundred 200-foot patrol boats for the U. S. Navy, on a cost plus fixed profit basis. The number of boats originally contracted for was reduced to sixty after the armistice was signed.

The site selected for the shipyard lies to the west of the city limits, about eight miles from the center of the city, near the River Rouge.

Active work in erecting buildings and dredging a slip communicating with the River Rouge was begun in March, 1918. This slip is about one-half a mile long, three hundred feet wide, and twenty-eight feet deep. The cost of the necessary buildings and equipment was about three and one-half million dollars.

The principal buildings erected were: The fabricating shop, the assembly building and store building. The transfer table, the hydraulic launching table and fit-out docks, constructed along with the above buildings, complete the plant equipment.

Plates, frames, shapes, etc., were manufactured in the fabricating shop, which connects with the assembly building.

The assembly building is 1,750 feet long and 300 feet wide. In it were laid three main tracks for specially made heavy trucks on which the hulls of the *Eagle* boats were built. As work progressed on a hull, it was moved on its trucks from one assembly stage to the next until it had passed through seven stages, when it was ready for launching. Each one of the three main tracks would accommodate a line of seven

hulls, making a total of twenty-one hulls under continuous assembly.

The transfer table is simply a truck about 190 feet long and 30 feet wide, which runs on heavy tracks parallel to the end of the assembly building. Motive power is furnished by a 30 horsepower motor.

When a hull has passed through its last stage of assembly in the assembly building, it is hauled on its own trucks out on to the transfer table, on which it is transported to a point opposite the hydraulic launching table. The hull, still on its own trucks, is then hauled on to the hydraulic launching table, lowered into the water, floats off its ways when water borne, and is then towed clear. The launching table is then raised, the trucks placed on the transfer table, and returned to the first stage in the assembly building.

The launching table is an hydraulic lift 190 feet in length and operated by eight hydraulic engines, four on each side. The capacity of the table is about 600 tons. The total time required to launch a hull, starting from the last stage in the assembly building, is about 40 minutes.

The *Eagle* boat is a single-screw vessel, driven by a high-speed Poole turbine through a planetary reduction gear, and is designed for a speed of 18 knots at about 500 tons trial displacement with the engine developing about 2,000 shaft horsepower.

GENERAL DIMENSIONS.

Length between perpendiculars, feet	200
Length over all, feet and inches	200–9
Beam molded at mid perpendicular, feet and inches	25–6
Depth molded from stem to bulkhead No. 59, feet and inches	18–6
Depth molded at after perpendicular, feet and inches	14–6
Mean trial displacement, tons (about)	500
Mean draft to bottom of keel at mean trial displacement, feet and inches	7–3
Fuel oil, approximate full load capacity, tons	90
Fuel oil, approximate emergency capacity, tons	145
Frame spacing, inches	21

Fuel oil is carried in six "D" compartments and three "A" compartments. Fresh water for feed and ship's use is carried in three tanks between bulkhead No. 59 and No. 65 in "C" compartment.

Amount of feed water carried, gallons	5,680
Amount of ship's water carried, gallons	2,400

COMPLEMENT.

Commanding officer	1
Wardroom officers	3
Crew, including chief petty officers	54
	58

BATTERY.

Two four-inch 50 caliber rapid fire rifles.

One three-inch 50 caliber anti-aircraft rifle, high power, flat trajectory type.

One Y gun, depth charge projector.

Two machine guns.

Depth charge release gear is intalled on port and starboard sides of after part of main deck.

SMALL BOATS CARRIED.

One 21-foot motor dory.

One 24-foot whale boat.

The boats are stowed on heavy curved channel davits. The davits rig in and out by means of a sheathed screw gear operated by a crank.

MACHINERY.

(See Plate VII for arrangement of machinery space.)

The main engine is a Poole high-speed turbine of the compound impulse velocity stage type, with a working pressure of 235 pounds. It is designed to develop 2,000 shaft horsepower at 4,000 revolutions per minute and is geared down in the ratio of 8.4 to 1.

U.S.S. EAGLE 1 TO 60
ARRANGEMENT OF MACHINERY IN ENGINE ROOM
PLATE VII
1 MAIN ENGINES
2 CONDENSER
3 AIR PUMP
4 CIRCULATING PUMP
5 MAIN CIRC. PUMP
6 MAIN FEED PUMPS
7 FEED FILTER
8 BILGE PUMPS
9 FRESH WATER DISTILLING APPARATUS
10 DISTILLING CIRC. PUMP
11 DISTILLING FEED PUMP
12 DISTILLING CONDENSER
13 LUBE OIL PUMPS
14 LUBE OIL COOLER
15 AUX. ENGINES
16 AUX. CIRC. PUMPS
17 AUX. FEED PUMPS
18 AUX. FEED FILTERS
19 AUX. FEED TANK
20 AUX. OIL TANK HEATER
21 AUX. OIL TANK
22 FEED WATER HEATER
23 BOILER
24 FEED PUMPS
25 SETTLING TANK
26 FUEL OIL SERVICE TANK

There are six pressure stages in the ahead turbine—the first or high-pressure stage is velocity compounded. Incorporated in the same casing and at the after end is the backing turbine, which consists of one pressure stage, velocity compounded. It has about 30 per cent of the power of the ahead turbine.

(For details of rotor see Plate VIII.)

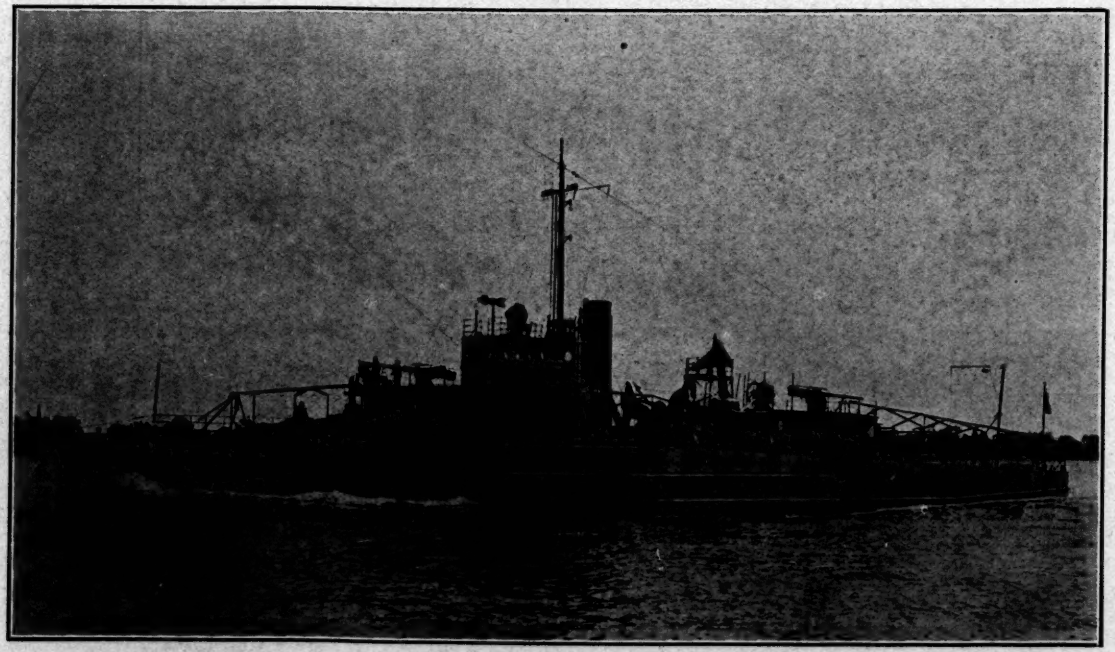
The rotor wheels are assembled in the central part of the shaft, keyed in place, distance pieces being placed between the wheels to maintain the positions. A nut on each end of the central portion of the shaft is screwed hard against the first and last wheels. These securing nuts are locked in place by set screws.

The first rotor wheel in the ahead turbine and the astern rotor wheel are of the same size and construction. Each carry two sets of blades.

The rotor shaft revolves in self-aligning bearings of the spherical seat type. The forward bearing is located in the forward end of the turbine casing, and the after bearing in the forward end of the reduction gear casing.

The central part of the shaft is 27 inches long and 6.998 inches in diameter. The shaft decreases in size on both sides of this central part to 5½ inches in diameter for lengths of 9¾ inches. These parts of the shaft provide smooth cylindrical surfaces, which revolve in a series of radial slotted projections forming the shaft gland packing. Sealing steam from the auxiliary exhaust line is admitted to an annular space at the center of the packing. The leak-off from the shaft packing at the high-pressure end is piped to the annular space in the low-pressure packing.

The turbine rotor is equipped with a small thrust bearing at the forward end, which takes up the thrust of the rotor shaft and maintains proper clearances between moving and stationary parts. The side clearance between moving and stationary parts is adjusted by means of distance pieces behind the thrust blocks. The after end of the rotor shaft is tapered



"EAGLE" No. 57.

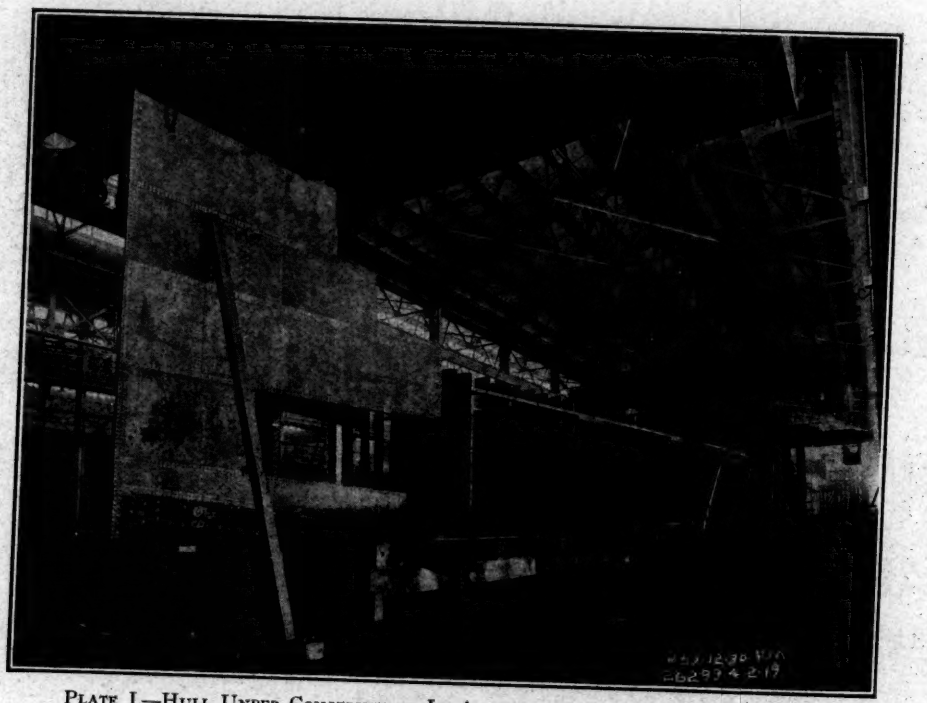


PLATE I.—HULL UNDER CONSTRUCTION IN ASSEMBLY BUILDING, SHOWING TRUCKS.

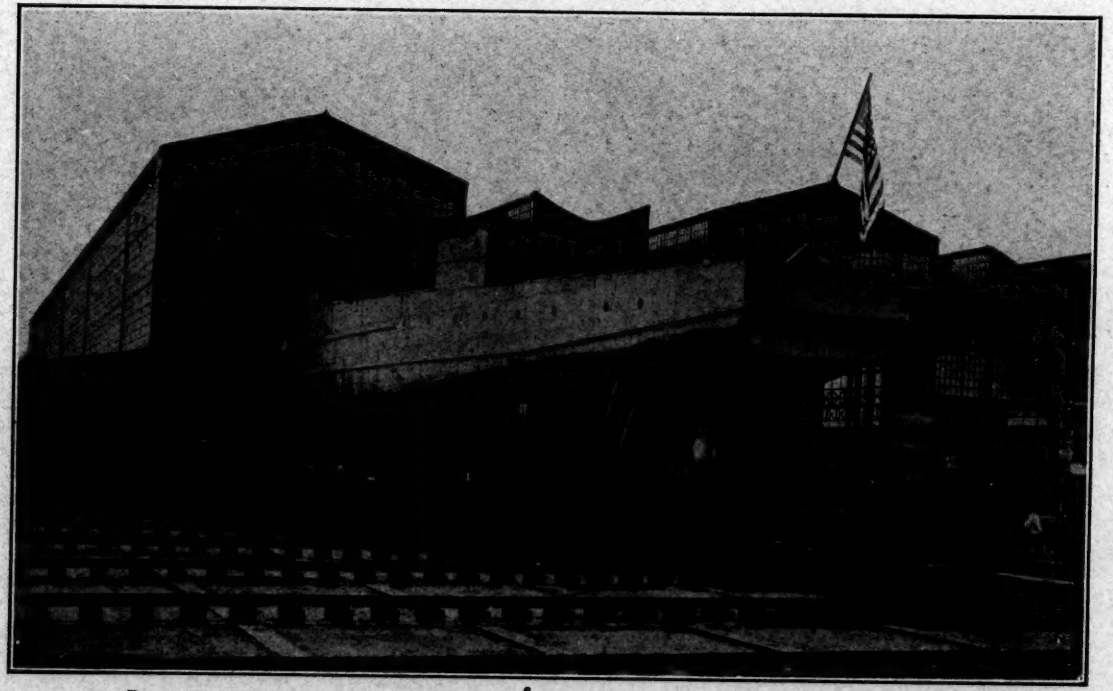


PLATE II.—HULL LEAVING ASSEMBLY BUILDING AND BEING HAULED ONTO TRANSFER TABLE.

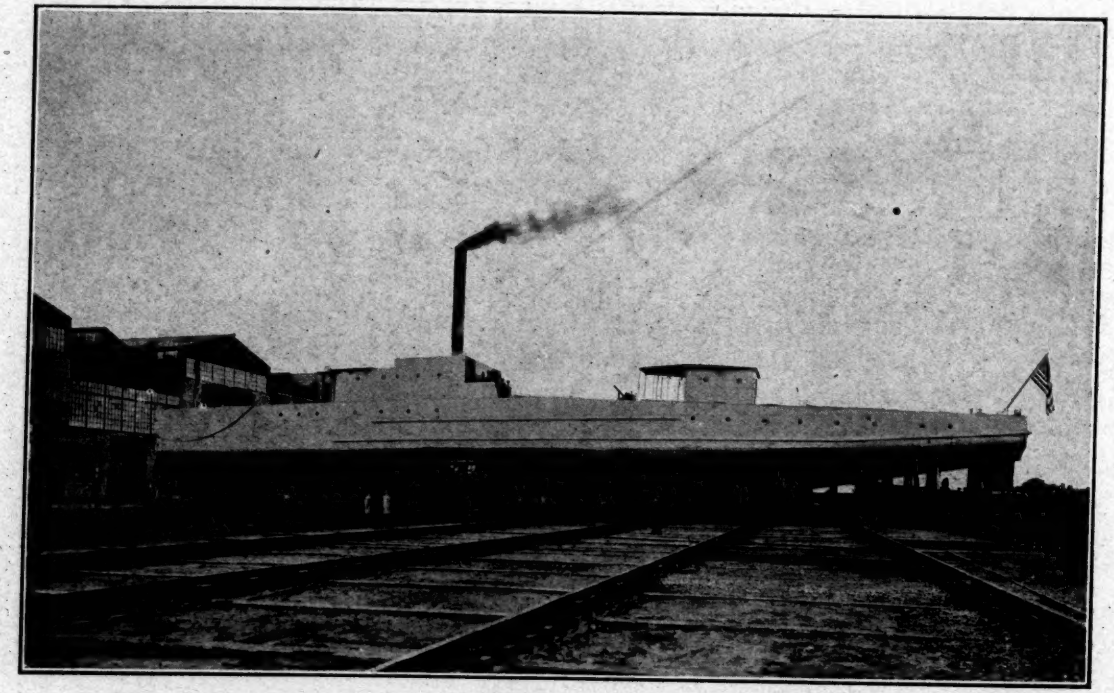


PLATE III.—HULL ON TRANSFER TABLE.

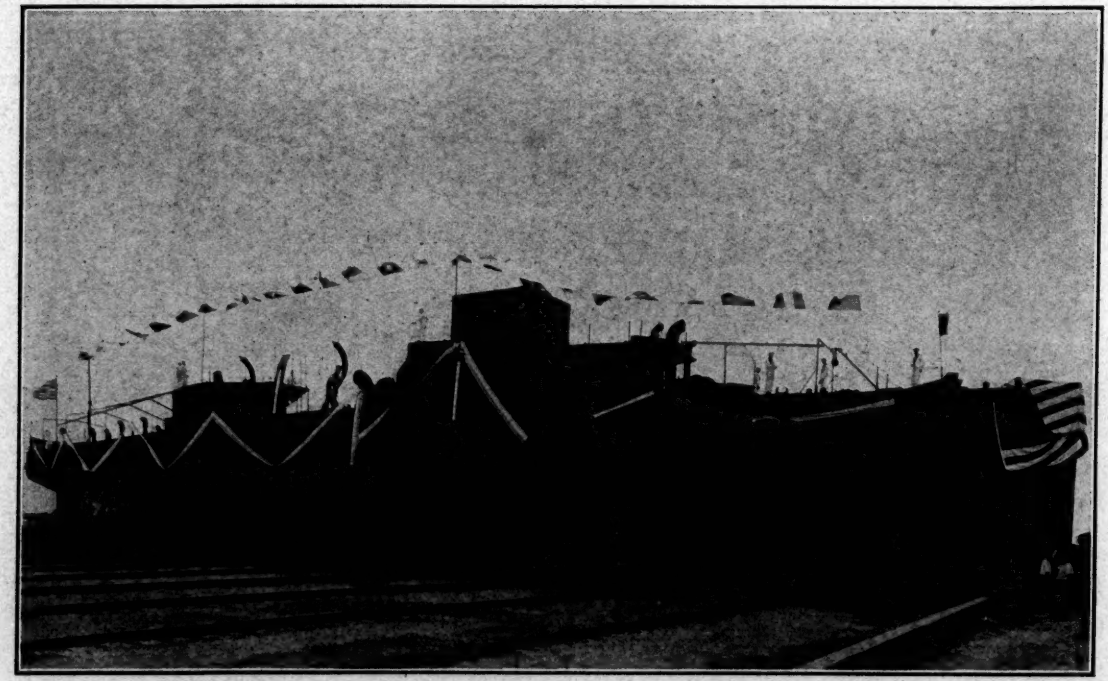


PLATE IV.—HULL ON TRANSFER TABLE.

PLATE V.—HULL ON LAUNCHING TABLE.

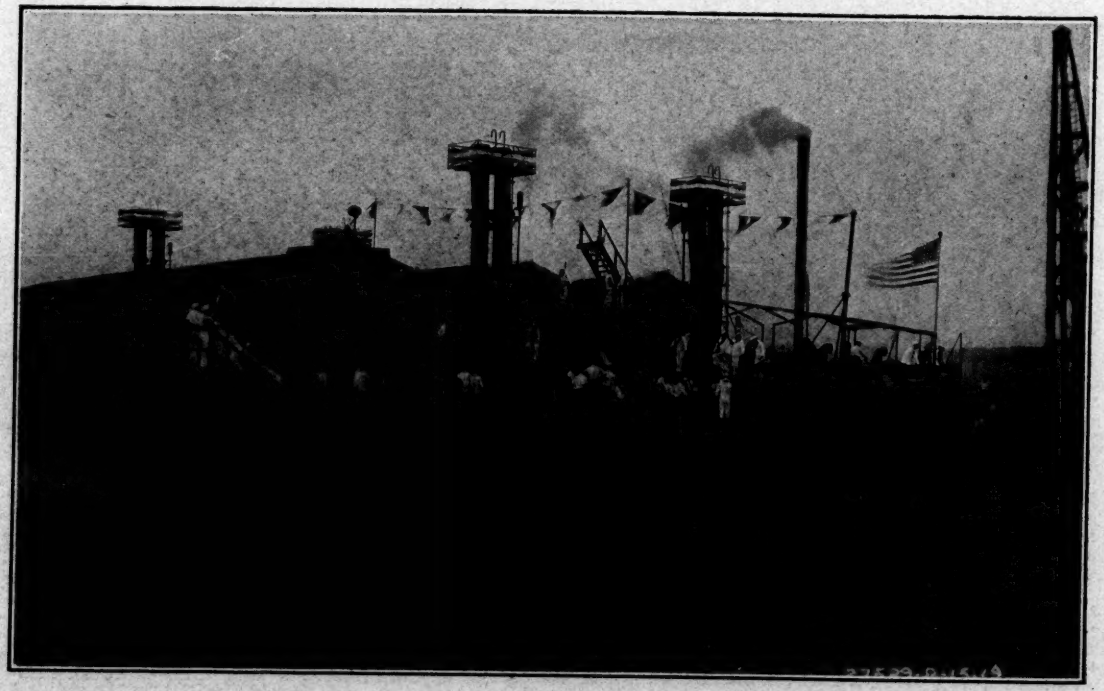


PLATE VI.—HULL FLOATING OFF LAUNCHING TABLE.

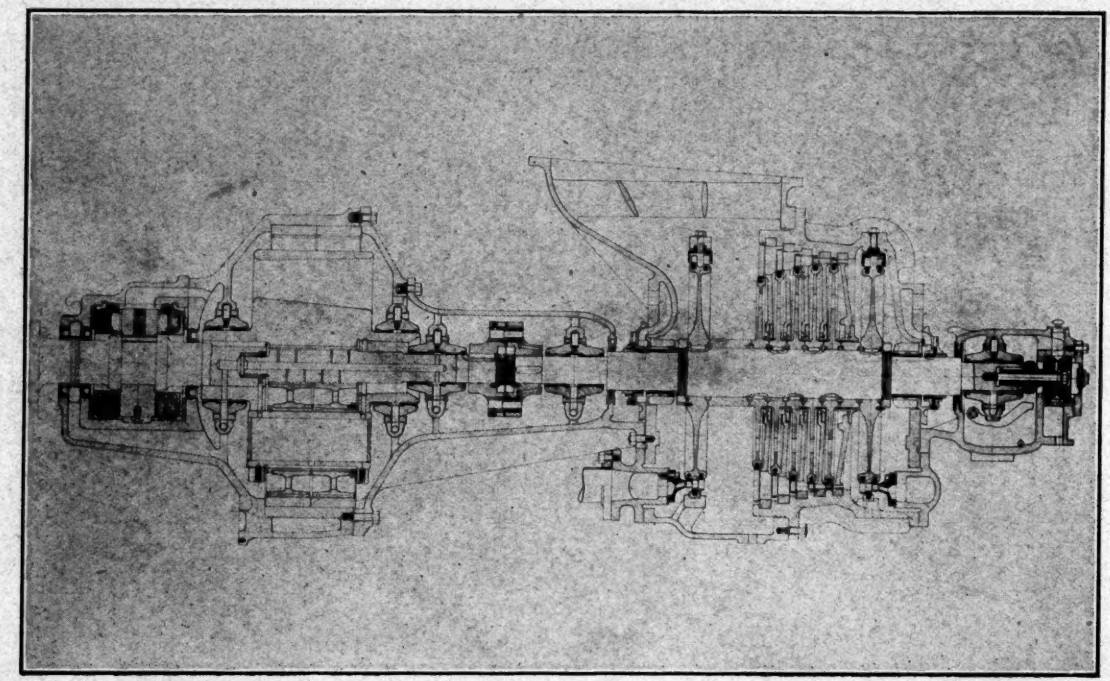


PLATE VIII.—ROTOR.

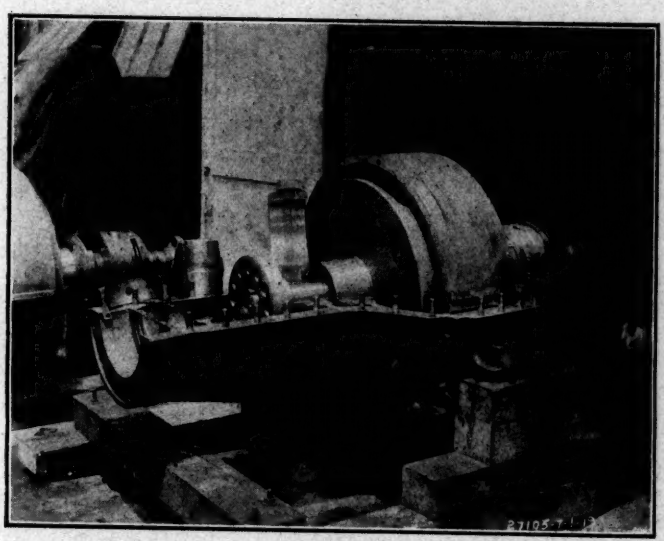


PLATE X.—REDUCTION GEAR.

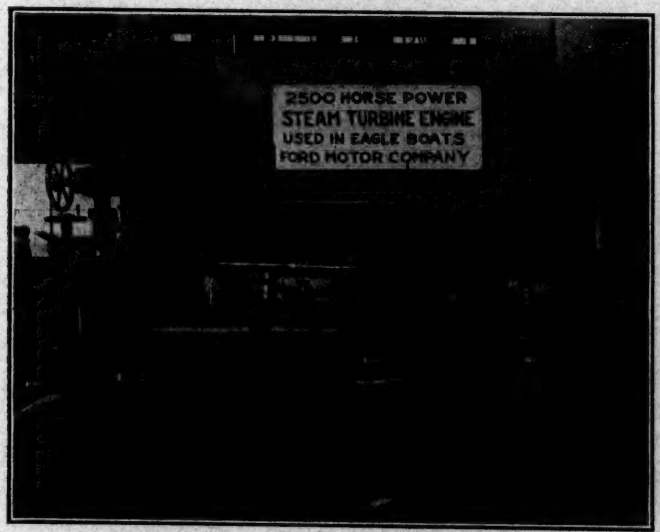


PLATE XI.—MAIN ENGINE ASSEMBLY.

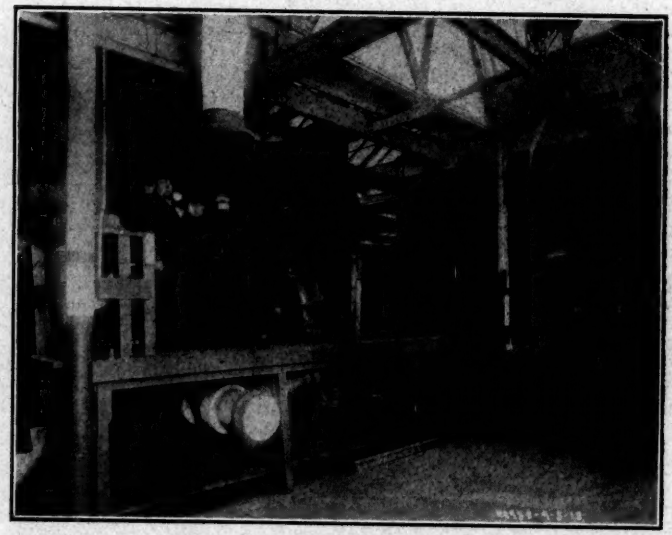


PLATE XII.—TURBINE TEST BLOCK.

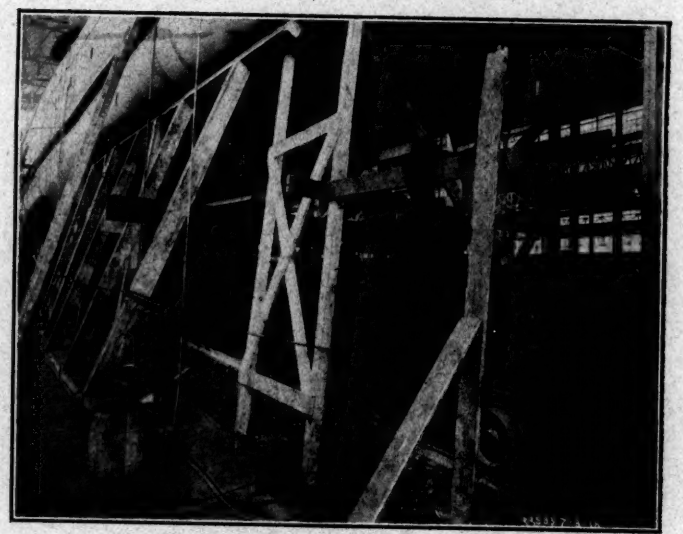


PLATE XIII.—BORING STERN AND STRUT CASTINGS.

PLATE XV.—EAGLE BOAT BOILER.

and threaded, and fitted with a nut with a key-way, for securing the linear expansion coupling flange.

The forward end of the reduction gear sun pinion shaft is of the same construction and has a corresponding linear expansion coupling flange. These coupling flanges are connected by twelve steel pins which are secured in the reduction gear flange, and which work in bronze sleeves fitted in the rotor flange. A space of 3/16 inch is left between coupling flange faces, thereby allowing for linear expansion of the rotor shaft and reduction gear sun pinion shaft.

(For details of blading see Plate IX.)

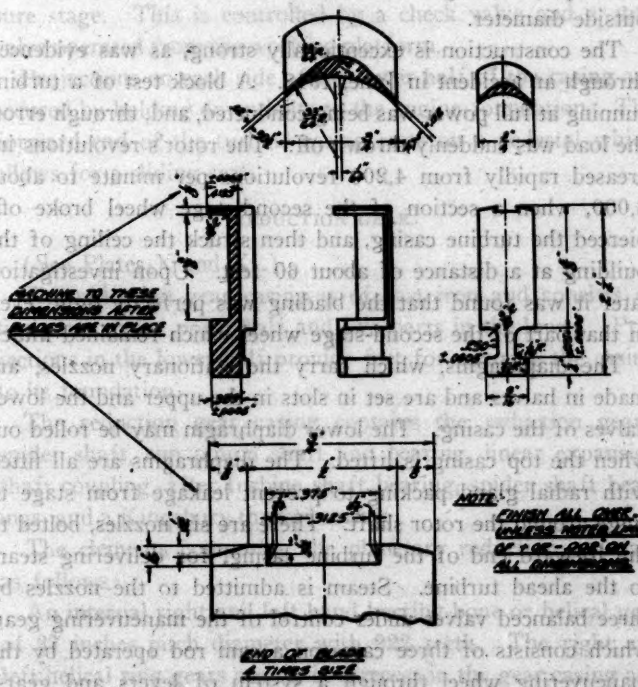


PLATE IX.—TURBINE BLADING.

The periphery of the rotor wheels are ⊥ slotted. The roots of the rotor blades are of a corresponding cross-sectional shape. A starting place for inserting the roots of the blades is provided, and, after all blades are inserted in the rotor slot, a locking piece is inserted and electrically welded.

The distance between the blades is maintained by the shapes of their roots and tips. After all the blades are assembled on a wheel, a dove-tailed slot is cut in their periphery and a flat strip of soft steel, in lengths of not over 10 inches, is then peened into this slot and electrically welded. This serves as a shroud ring. The tips of the blades are then ground to smooth them up and bring the assembled rotor wheel to its finished outside diameter.

The construction is exceptionally strong, as was evidenced through an accident in June, 1918. A block test of a turbine running at full power was being conducted, and, through error, the load was suddenly thrown off. The rotor's revolutions increased rapidly from 4,200 revolutions per minute to about 8,000, when a section of the second-stage wheel broke off, pierced the turbine casing, and then struck the ceiling of the building at a distance of about 60 feet. Upon investigation later it was found that the blading was perfectly tight, even in that part of the second-stage wheel which remained intact.

The diaphragms, which carry the stationary nozzles, are made in halves and are set in slots in the upper and the lower halves of the casing. The lower diaphragm may be rolled out when the top casing is lifted. The diaphragms are all fitted with radial gland packing to prevent leakage from stage to stage around the rotor shaft. There are six nozzles, bolted to the forward end of the turbine casing, for delivering steam to the ahead turbine. Steam is admitted to the nozzles by three balanced valves under control of the maneuvering gear, which consists of three cams on a cam rod operated by the maneuvering wheel through a system of levers and gears. Each valve admits steam to two nozzles. Steam is admitted

to the astern turbine nozzles by a 4-inch double-seated balanced valve under control of the maneuvering gear.

A double-seated balanced steam-stop valve, operated from the working platform, is fitted between the steam chest and the main steam line. A foot trip gear is installed so that this valve may be closed quickly in case of emergency. A centrifugal ring governor is installed on the forward end of the rotor shaft. This operates a trip gear which disengages a ratchet that holds the stop valve open and allows the valve to close when the revolutions of the rotor exceed a predetermined number.

Auxiliary exhaust steam may be admitted to the first pressure stage. This is controlled by a check valve and a stop valve operated from the working platform.

Projections on each side of the lower half of the casing are secured by holding down bolts to the engine foundation. The forward end of the turbine casing rests on a pedestal which allows for a sliding seat.

REDUCTION GEAR.

(See Plates X and XI.)

The reduction gear casing is of cast iron and consists of five parts in the upper half and two parts in the lower. Projections in the lower half provide feet for securing the casing to its foundation.

The reduction gear casing contains the reduction gears, spider shaft, sun pinion shaft and bearing, linear expansion shaft coupling, after turbine shaft bearing, spider shaft bearings, and a Kingsbury thrust bearing.

The elements composing the planetary reduction gear are as follows:

An internal right and left hand herring-bone or helical gear of 37 inches pitch diameter with 222 teeth. The right and left helical ring gears are set in a recess in the gear casing and keyed together and in place.

Revolving inside and in mesh with the internal ring gear are three planetary gears with the same type teeth as the ring gear. These planetary gears are 16 inches pitch diameter and have 96 teeth. They revolve on pin journals, carried on a revolving steel spider.

The planetary gears also mesh with a central sun pinion right and left hand helical gear whose shaft carries the after expansion coupling at its forward end. The forward spider bearing which is 10 inches in diameter is hollow in order to allow the sun pinion shaft to pass through.

The sun pinion shaft is connected to the after end of the main turbine shaft by the expansion coupling. The sun pinion gear, revolving the planetary gears around their pins and the internal ring gear, causes the planetary gears to give the spider with its shaft a reduction in the number of revolutions.

A Kingsbury thrust collar is fitted on the after end of the spider shaft; the thrust blocks are set in the after part of the reduction gear casing.

All bearings of the turbine and reduction gear are lubricated by the forced feed system. A pressure of about 40 pounds is carried on the after 10-inch spider shaft bearing and the 4-inch pinion shaft bearing. The high-pressure oil lubricates the 4-inch pinion shaft bearing and then passes through a hole in the shaft to the pinion gear. Oil holes are drilled in the pinion gear to communicate with the oil hole in shaft, and the lubricating oil is distributed to all the gear teeth from the pinion gear. High-pressure oil from the after 10-inch spider bearing passes through the spider shaft journal to oil passages in the spider which leads to the oil chambers in pin journals of the planetary gears.

Lubricating oil at about 6 pounds pressure is led to all other bearings.

All main turbines and reduction gears were manufactured at the Ford Motor Company's Highland Park Plant. As each turbine and gear was completed, it was assembled as a

unit and put on a test block and given a running test, at least four hours of which was at full power. The load was taken by a 1,500 kilowatt motor generator. The main engine auxiliaries—condenser, condensate pump, radojets, lubricating oil pumps and lubricating oil coolers—were used in connection with this test. Plate XII shows this installation. (For details of block test see Table I.)

TURBINE DATA.

Number of rotor wheels in ahead turbine	6
Number of rotor wheels in astern turbine	1

First Wheel.

510 blades, blade opening, inches	1.375
Width of blade, inch	.75
Length over all, inches	3.21875
Thickness of shrouding, inch	.413
Thickness of blade base, inch	.359

Second Wheel.

217 blades, blade opening, inch	.5625
Width of blade, inch	.75
Length over all, inches	1.4375
Thickness of shrouding, inch	.474
Thickness of blade base, inch	.433

Third Wheel.

217 blades, blade opening, inch	.875
Width of blade, inch	.75
Length over all, inches	1.84375
Thickness of shrouding, inch	.529
Thickness of blade base, inch	.473

Fourth Wheel.

217 blades, blade opening, inches	1.5625
Width of blade, inch	.75
Length over all, inches	2.4375
Thickness of shrouding, inch	.488
Thickness of blade base, inch	.418

TABLE I.

Power Department. FORD MOTOR COMPANY. Highland Park Plant.

Steam Turbine Test.

No. 2 turbine fitted with one-inch nozzles instead of regular 5/8-inch nozzles as no high-pressure steam was available.

Date	July 16, 1918, A.M.					
Hour	4—5	5—6	6—7	7—8	8—9	9—10
R.P.M.	4,320	4,320	4,320	4,320	4,320	4,320
E.H.P.	1,980	2,006	1,985	2,042	2,291	2,490
Steam pressure (gauge)	154.5	153.5	151.0	154.0	166.0	176.0
Steam temperature (degrees F.)	396.0	399.0	403.0	403.0	409.0	424.0
Superheat (degrees F.)	27.5	31.5	36.5	35.7	35.2	46.1
Barometer	29.32	29.32	29.32	29.30	29.30	29.30
Vacuum (inches of mercury)	27.5	27.4	27.4	27.39	27.26	27.20
Absolute pressure in condenser (inches of mercury)	1.82	1.92	1.92	1.91	2.04	2.10
Water condensed per hour (pounds)	25,500	25,800	25,500	25,800	29,400	29,880
Water condensed per e.h.p. hour	12.85	12.72	12.85	12.60	12.80	12.01
Oil used	Ursa.	Ursa.	Ursa.	Ursa.	Ursa.	Ursa.
Total oil to turbine and gears (pounds per hour)	29,300	29,550	29,200	29,900	29,200	28,410
Oil to turbine forward bearings (pounds per hour)	8,400	8,200	8,400	8,500	8,200	8,200
High pressure to gears only (pounds per hour)	13,150	13,250	13,400	13,600	13,000	12,800
Total oil less forward bearing (pounds per hour)	20,900	21,350	20,800	21,400	20,700	20,210
Temperature of oil to gears and bearings (degrees F.)	132.0	132.0	132.0	134.0	134.5	135.0
Temperature of oil from gears and bearings (degrees F.)	157.0	157.5	158.0	158.0	157.5	158.75
Temperature rise through gears and bearings	25.0	25.5	26.0	24.0	23.0	23.75
B.T.U. per hour absorbed by gears and bearings except forward bearing	260,400	271,100	270,100	257,000	234,000	218,000
H.P. loss—Gears and bearings (except forward bearing)	103.5	105.2	105.0	100.0	90.6	84.5
Efficiency of gears and bearings	95.0	95.25	95.5	95.5	96.25	97.0
Weight of oil through coolers	29,450	29,700	29,350	30,050	29,350	28,560
Temperature of oil to coolers (degrees F.)	152.0	152.0	152.5	153.0	153.8	154.0
Temperature of oil from coolers (degrees F.)	132.0	132.0	132.0	134.0	134.5	135.0
Temperature drop through coolers (degrees F.)	20.0	20.0	20.5	19.0	19.3	19.0
B.T.U. absorbed in coolers per hour	294,000	297,000	300,000	286,000	279,000	273,000
Quantity water through coolers (pounds per hour)	59,500	61,900	60,050	56,100	56,250	59,500
Temperature of cooling water	66	66	66	66	66	66
Room temperature	81	80	81	81.5	81.5	82.3

Note.—150 pounds of oil per hour to pedestal bearings added on for oil coolers.
Turbine direct connected to motor generator set.

/s/ W. F. Kendrick. 7/19/18.

Fifth Wheel.

217 blades, blade opening, inches	3.0
Width of blade, inch	.75
Length over all, inches	3.875
Thickness of shrouding, inch	.509
Thickness of blade base, inch	.397

Sixth Wheel.

243 blades, blade opening, inches	4.625
Width of blade, inch	.75
Length over all, inches	5.5
Thickness of shrouding, inch	.476
Thickness of blade base, inch	.334

In the periphery of the rotor wheel is a tee slot in which the blades are secured.

DIAPHRAGMS.

Second Stage.

56 blades—	
Pitch of blades, inches	1.79
Diameter of diaphragm, inches	34.498

Third Stage.

56 blades—	
Pitch of blades, inches	1.79
Diameter of diaphragm, inches	34.748

Fourth Stage.

56 blades—	
Pitch of blades, inches	1.79
Diameter of diaphragm, inches	35.498

Fifth Stage.

56 blades—	
Pitch of blades, inches	1.79
Diameter of diaphragm, inches	36.998

Sixth Stage.

48 blades—	
Pitch of blades, inches	2.094
Diameter of diaphragm, inches	38.498

All diaphragms are bored to receive packing ring 11½ inches outside diameter.

REDUCTION GEAR DATA.

Material.

Housing enclosing gear	Cast iron.
Spider	Steel casting.
High speed pinion shaft	Chrome Vanadium steel.
Planetary gears	Chrome Vanadium steel.
Gear ring	Chrome Vanadium steel.
Low speed shaft	Chrome Vanadium steel.
Shaft couplings	Chrome Vanadium steel.
Planetary gear pin	Chrome Vanadium steel.
Coupling pins in expansion coupling	Chrome Vanadium steel.
Oil glands	Bronze.

Gears.

Internal gear, teeth	222
Planetary gears, teeth	96
Pinion gear, teeth	30
Internal gear, pitch diameter, inches	37
width, inches	6
Planetary gear, pitch diameter, inches	16
width, inches	6
Pinion gear, pitch diameter, inches	5
width, inches	6
Helix angle of teeth	230
Ratio of reduction	8.4 to 1

All bearings in turbine and reduction gear are of the self-aligning type setting on spherical seats.

RATES OF TURNING.

Turbine shaft, revolutions per minute	4,000
Sun pinion gear, revolutions per minute	4,000
Planetary gear on pins, revolutions per minute	1,735
Line shaft and propeller shaft, revolutions per minute	475

SHAFTING LINE.

Two sections of line shaft, length, feet and inches	13–3
One section of tail shaft, length, feet and inches	28–9⅝
Total length of shafting, feet and inches	55–5⅝
Rake per foot of shafting, inch	.849
Diameter of line shafts, inches	6.25
Diameter of tail shaft, inches	6.375

There is one right hand, cast solid, manganese bronze, 3-bladed propeller, machined to pitch and polished. Diameter, 6 feet 7 inches; true pitch, 4 feet 10 inches.

SPRING BEARINGS.

There are two spring bearings supporting the line shafting. Size of bearing, 6¼ inches by 12 inches. Lined with white metal.

LIGNUM VITAE BEARINGS.

Lignum vitae bearings are fitted at frames No. 92 and No. 106.

Lignum vitae strips, 11/16 × 2 feet 3 inches.

Bearings bored out 1/32 larger than shaft for clearance.

Zincs are fitted on both ends of strut bearings and after end of stern tube.

(See Plate XIII for method of boring stern and strut castings.)

Strut and stern castings are bored by a specially constructed, motor-driven, boring machine mounted on wheels, which run on the assembly track. Witness marks for the alignment of the boring bar are secured by reference to a weighted calibrated steel wire placed in the position of the center line of the propeller and line shafting.

MAIN CONDENSERS.

There is one main condenser of the cylindrical, straight-tube type. In the lower half of its forward end and in the upper half of its after end, the tubes are made tight by being expanded into the tube sheets. The opposite ends of the tubes are made tight by means of the usual stuffing box, corset lacing packing and ferrules.

The principal dimensions are as follows:

Distance between tube sheets, feet	7
Length of tubes as fitted, feet and inches	7–2½
Thickness of tube sheets, inches	1⅛

Tubes, number	2,084
Tubes, diameter outside, inch	⅝
Tubes, thickness, B.W.G.	16
Cooling surface, square feet	2,387
Length of condenser, feet and inches	9–8
Inside diameter, feet and inches	4–10

The condenser has the following connections on the fresh water side:

One main exhaust nozzle rectangular in shape;
One condensate pump suction;
One air ejector suction;
One auxiliary exhaust steam connection;
Trap drains;
One safety valve, water sealed, inside diameter 1½ inches;
One hand hole and cover;
One vent connection from condensate pump;
Vacuum gauge connection;
Bleeder connection;
Soda cock;
Vacuum line connections for revolution counter and quick closing stop valve pilot valve.

And the following connections on water chest:

One main injection connection on forward water chest;
One overboard discharge connection on forward water chest;
One drain on each chest;
One thermometer connection on injection piping;
Air cocks.

Vacuum is established and maintained in the main condenser by means of a turbo-driven, centrifugal condensate pump and two Wheeler radojets. The condensate pump is direct connected to a 4-horsepower Terry steam turbine of the simple impulse type. This turbine has a fly ball type governor which cuts off and admits steam according to the load. In addition, there is an emergency trip governor which closes the steam admission valve in case the speed of the turbine exceeds safe limits. The condensate pump is capable of de-

livering 30,000 pounds condensate per hour against a head of 60 feet at 3 horsepower and making 2,500 revolutions per minute. The maximum capacity of this pump is 45,000 pounds condensate per hour.

The radojets remove air from the condenser by the suction effect created by a series of steam jets. They consist of two steam ejectors working in series; the upper and the lower ejectors constituting the first and second stages, respectively.

In operation live steam from a reducing valve is delivered to the upper or first series of jets. The steam expands through these jets or expansion nozzles and gains a high velocity. The jets of steam from the first stage pass across a suction space, in communication with the main condenser, where they mix and draw air from the condenser with them.

The mixture of air and steam enters a large diverging nozzle called "the diffuser," from where it is discharged at a higher pressure than at the original suction space, into a double-passage chamber surrounding the second series of jets. The steam entering the second series of jets expands radially and leaves at a high velocity and collects the mixture in the double chamber compressing it above atmospheric pressure in an annular chamber from which it is discharged past a check valve to the feed and filter tank.

When warming up the main engine and when "standing by," the temperature of the water in the feed tank is kept under the boiling temperature by a recirculating line with a thermostatic valve.

When the water in the feed tank reaches a temperature below 212 degrees F. and for which the thermostatic valve is set, the valve opens and the partial vacuum in the condenser causes the water from the feed tank to flow through the recirculating line into the main condenser where it is cooled by the main condenser circulating water. The condensate pump discharges the water back into the feed tank along with the condensed steam.

A turbo-driven double inlet centrifugal pump supplies circulating water to the main condenser. The circulating pump is arranged to take its suction from the sea through the main injection casting or, by closing the main injection valve, the bilge suction valve may be opened and injection water taken from engine room bilge.

The main circulating pump is direct connected to a type Z N 33 horsepower Terry steam turbine and is capable of discharging 3,600 gallons of water per minute against a head of 25 feet when making 1,300 revolutions per minute.

AUXILIARY CONDENSER.

There is one 20-inch diameter cylindrical shell surface condenser, containing 225 square feet of cooling surface. The condenser is mounted upon a direct acting combined air and circulating pump, having a 5-inch steam, a 6-inch air, and a 6-inch water cylinder. There is a spring-loaded valve on the auxiliary exhaust line to the condenser for regulating the back pressure.

FEED AND FILTER TANK.

The feed and filter tank is located in the forward starboard side of the engine room. It is built of steel plate 3/16 inch thick. A perforated plate separates the filter tank from the feed tank proper. The filter tank has the following connections: One condensate pump discharge, one air ejector discharge, drain pipes from coils and traps. The feed tank has the following connections: Feed suction, drain line from auxiliary machinery, drain cock, vapor escape, water gauge glass fittings, boiler compound connection to compound tank, and thermometer.

FRESH WATER SYSTEM.

The ship's tanks have a capacity of 2,400 gallons. Filling lines are installed from ship's side and from the distiller fresh-

water pump. Water is supplied to gravity tanks by a 4-inch horizontal hand pump located in the engine room.

A 75-gallon gravity tank is located in the galley. Branches run to a faucet for galley use and to a faucet outside the galley for bucket use. Officers' state rooms are provided with running water from the galley fresh-water tank. There is a 25-gallon gravity tank in the crew's work room aft.

MAIN AND AUXILIARY FEED SYSTEM.

There are two Blake-Knowles simplex, vertical, double-acting, reciprocating main feed pumps (9 inches × 6 inches × 12 inches), secured to a heavy standard in the forward starboard side of the engine room. The feed pumps take their suction from either the feed tank, the reserve feed tank, the fresh-water tank, or main condenser through a double-suction manifold. The steam lines to the pumps are fitted with Ideal pressure governors, and there is a spring-loaded relief valve on the discharge chamber of each pump.

The discharge lines from the pumps connect into a single 3-inch copper line, which leads to a No. 12 Reilly heater secured to the forward engine room bulkhead. (This heater is capable of heating 31,000 pounds of water per hour from 80 degrees F. to 215 degrees F., using exhaust steam at 10 pounds, per gauge.) The main feed line passes from the feed heater through the forward engine room bulkhead into the fire room. Here the feed line branches into two 3-inch lines, with a 3-inch high pressure gate valve on each branch. The right hand is the main feed line, and from a tee it branches into two 2-inch lines which lead to the main feed checks. From a Y on the left 3-inch line, two 2-inch lines lead to the auxiliary feed checks.

LUBRICATING OIL SYSTEM.

The forced feed lubricating oil system consists of following equipment:

One oil storage tank of 250 gallons capacity;
Four oil return tanks of about 114 gallons each;
One oil treating tank, three sections, fitted with heating coils, filtering cartridges and a settling space;
Two Kinney turbo-driven rotary oil service pumps;
One twin oil strainer in discharge piping;
One strainer of the McComb type in the suction piping to service pumps;
Two vertical multi-whirl oil coolers;
One 4½-inch × 5-inch × 6-inch single vertical double-acting oil cooler pump;
Electric flometers on each discharge lead to bearings.

If oil stops flowing for any reason, the trouble is indicated by a light on the gauge board. Each discharge lead to bearings is fitted with a Pitot tube which shows the volume and the pressure of the oil flowing in the pipe.

BOILERS.

There are two oil-burning boilers of the Bureau of Steam Engineering express type, arranged in pairs in one compartment, as shown on Plate XIV. Plate XV shows a boiler without casing or fittings. Nearly all of these boilers were built at the Ford Motor Company's Highland Park Plant. These boilers were assembled with their casings, brickwork and fittings on the fit-out dock. They were then picked up with a locomotive crane and lowered into place.

Each boiler has 24 rows of tubes, 12 on each side. The first two rows of tubes next to the combustion chamber are 1⅜ inches outside diameter and the others are 1⅛ inches outside diameter. There are 114 of the former and 950 of the latter. There is a fore-and-aft division plate inside the smoke pipe extending from the uptake to within a few feet of the top of the pipe. This is necessary in order that work may be done in one boiler while steam is being kept up in the other.

1 - BOILERS
2 - FORCED DRAFT BLOWERS
3 - FUEL OIL SERVICE PUMPS
4 - FUEL OIL BOOSTER PUMP
5 - FUEL OIL STAND-BY PUMP
6 - FUEL OIL HAND PUMP
7 - FUEL OIL HEATER
8 - AIR COMPRESSOR
9 - PORTABLE FIRE EXTINGUISHER
10 - CONDENSSOR AIR TANK

PLATE XIV

U.S.S. EAGLE 1 TO 60

ARRANGEMENT OF MACHINERY IN BOILER ROOM

BOILER DATA.

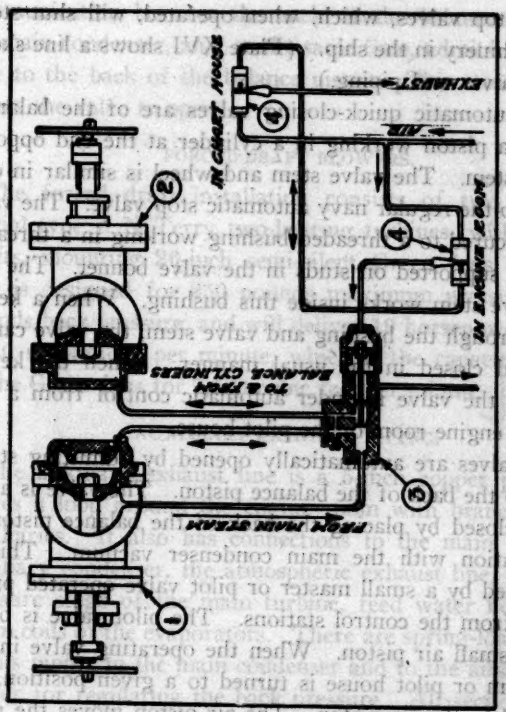
Number	2
Working pressure, pounds	250
Height to top, external, feet and inches	10–1½
Length at floor plates, feet and inches	7–8½
Width at floor plates, feet and inches	9–10
Number of furnaces to each boiler	One.
Furnace volume—each boiler, cubic feet	163
Capacity when full, water at 80 degrees, gallons	778
at steaming level, gallons	566
Heating surface—each boiler, square feet	1,500
Area through smoke pipe, square feet	9.6
Kind of draft	Closed fire room.
Atomizers—number each boiler	3
Diameter of main stop, inches	3½
auxiliary stop, inches	2½
main feed stop and check, inches	2
auxiliary stop and check, inches	2
surface blow valve, inches	1¼
bottom blow valve, inches	1½
safety valve (two) duplex—each, inches	2½
Height of funnel above floor plates, feet and inches	43–2¾
Air pressure carried at full power, inches	8
Steam space, each boiler, cubic feet, about	32
Water surface in each drum at steam level, square feet, about	25½
Type of atomization	Navy standard, pressure.

MAIN AND AUXILIARY STEAM PIPING.

Main steam piping is seamless drawn steel 203 mils thick: all joints are Van Stone. A 3½-inch main steam pipe from each boiler combines in the fire room into one 5-inch line which runs aft to the turbine. There is a bulkhead stop on this line in the engine room.

A 2¾-inch auxiliary steam pipe from each boiler combines into one 3¼-inch line in the fireroom which leads aft through the engine room bulkhead where it branches and makes a loop around the engine room. There is a stop valve on this line at the engine room bulkhead.

There is an automatic quick-closing stop valve on the main and on the auxiliary steam piping in the engine room just



U.S.S. EAGLE 1 TO 60
DIAGRAM OF QUICK CLOSING VALVE PLATE XVI.

abaft the stop valves, which, when operated, will shut steam off all machinery in the ship. (Plate XVI shows a line sketch of these valves and piping.)

The automatic quick-closing valves are of the balanced type with a piston working in a cylinder at the end opposite the valve stem. The valve stem and wheel is similar in construction to the regular navy automatic stop valve. The valve wheel is secured to a threaded bushing working in a threaded cross-piece supported on studs in the valve bonnet. The end of the valve stem works inside this bushing. When a key is inserted through the bushing and valve stem, the valve can be opened and closed in the usual manner. When the key is withdrawn the valve is under automatic control from a station in the engine room or the pilot house.

These valves are automatically opened by admitting steam pressure to the back of the balance piston. The valve is automatically closed by placing the back of the balance piston in communication with the main condenser vacuum. This is accomplished by a small master or pilot valve operated pneumatically from the control stations. The pilot valve is operated by a small air piston. When the operating valve in the engine room or pilot house is turned to a given position, air is admitted to the air piston. The air piston moves the pilot valve so that the steam pressure is cut off the back of the balance piston in the quick-closing valve. At the same time the pilot valve places the back of the quick-closing valve piston in communication with the main condenser. The partial vacuum back of the balance piston causes the valve to close. The valve remains closed as long as pneumatic pressure holds the pilot valve in this position. When it is desired to open this valve, the control valve at the operating station is placed in such a position that the air pressure behind the pilot valve air piston is released to atmosphere. A small steam piston which always has steam pressure on it now predominates and moves the pilot valve back to its original position, which cuts off

communication between the back of the balance piston and the main condenser, and, at the same time, admits steam pressure to the back of the balance piston. This steam pressure causes the valve to open.

FORCED DRAFT BLOWERS.

The forced draft installation consists of two horizontal special type Z N Terry two-bearing turbines, with extended shafts, mounting 26-inch semi-silent Green fans. The turbine is designed for 250 pounds maximum pressure and 10 pounds back pressure, and will deliver 19 horsepower at about 1,860 revolutions per minute, which is the capacity required by the Green fans for 7,500 cubic feet per minute.

AUXILIARY EXHAUST PIPING.

The auxiliary exhaust line is a 5-inch copper pipe which makes a loop around the engine room with branches to the auxiliaries. It also has connections to the main condenser, auxiliary condenser, the atmospheric exhaust line, first ahead pressure stage of the main turbine, feed water heaters, and steam coils of the evaporators. There are spring-loaded valves in this piping to the main condenser and to the auxiliary condenser for regulating the back pressure. All sections of this piping have Van Stone joints.

FUEL-OIL SYSTEM.

This system consists of one booster pump, two fuel-oil service pumps, one auxiliary fuel-oil service pump, a Reilly fuel-oil heater, valves and piping as required. The service pumps are the Kinney, turbo-driven, rotary plunger type. The fuel-oil suction piping is so arranged that the fuel-oil can be pumped from any one compartment to any other, or overboard, with the booster pump. The service pumps take their suction from the booster pump or direct from the fuel oil compartment. Suction piping has Van Stone joints. All pressure piping has flanged metal to metal joints.

The auxiliary fuel-oil service pump is a 4-inch × 2½-inch × 5-inch simplex, direct, double-acting, reciprocating pump, manufactured by the Union Steam Pump Company, installed on starboard side aft in fire room for port use. It has a connection to fuel oil suction manifold and discharges into the fuel oil discharge line.

AIR COMPRESSOR.

There is one vertical, steam-driven, direct-acting 11-inch × 11-inch × 12-inch Westinghouse standard air compressor installed on the port side of the fire room, with a 24⅜-inch × 78-inch air reservoir. Pneumatic lines lead from the reservoir through the machinery spaces with convenient connections for blowing boiler tubes, air atomization for galley range burners, piping to quick-closing stop valves and for operating pneumatic tools.

EVAPORATING AND DISTILLING PLANT.

This consists of:

Two high-pressure, single-effect Reilly evaporators of the vertical, submerged, multicoil type;

One vapor separator;

Two vertical, multicoil distillers;

One distiller test tank of 50 gallons capacity;

One 3½-inch × 4-inch × 4-inch distiller fresh-water pump;

One 7-inch × 7-inch × 12-inch distiller circulating pump;

One 3½-inch × 4-inch × 4-inch evaporating feed pump;

One evaporator feed water heater.

The rated capacity of the plant is 2,500 gallons per day. The system is provided with reducing valves on steam to coils, and traps to take care of coil condensate. Provision is made for supplying steam coils with steam from auxiliary exhaust piping when working at reduced capacity.

DATA.

One Evaporator.

Type	No. 6 Reilly, vertical, multicoil (submerged).
Diameter, inside, inches	22
Height over all, feet and inches	5–11⅛
Number of coils	6
Diameter of coils, inch	1
Thickness of coils	No. 16 B.W.G.
Heating surface, square feet	16.1
Diameter of steam nozzle, inches	1½
vapor nozzle, inches	2½
feed valve, inch	¾
blow valve, inches	3

One Distiller.

Type	"D" No. 5 Reilly, vertical, multicoil.
Diameter, inside, inches	16½
Height over all, feet and inches	4–3
Number of coils	5
Diameter of coils, inch	1
Thickness of coils	No. 16 B.W.G.
Cooling surface, square feet	19.85
Diameter of vapor inlet, inches	2½
Drain connection, inch	1
Circulating water connection, inches	2

Evaporator Feed Water Heater.

There is one type "D" No. 2 Reilly, vertical, multicoil feed water heater located on the engine room bulkhead above the evaporators.

Diameter, inside, inches	10
Number of coils	2
Height over all, feet and inches	4–3
Diameter of coils, inch	1
Thickness of coils	No. 16 B.W.G.
Heating surface, square feet	7.94
Diameter vapor inlet, inches	2
outlet, inches	2
Diameter feed inlet, inches	1¼
outlet, inches	1¼

REFRIGERATING INSTALLATION.

One Audiffren-Singrum, one-half ton, sulphur dioxide refrigerating machine with cold storage box is installed on *Eagles* Nos. 1 to 50, inclusive, in a compartment on the port side just forward of frame 50. The York, one ton C O^2, refrigerating machine is installed on *Eagles* Nos. 51 to 60, inclusive.

PNEUMERCATOR.

A pneumercator system to indicate the level of oil in the fuel oil compartments is installed in *Eagles* Nos. 8 to 56, inclusive.

STEERING GEAR.

A steering engine of the Williamson type is installed on the after engine room bulkhead. This steering engine is of the two-cylinder, differential valve type, operating at 125 pounds pressure.

The steering gear is of the right and left screw gear type. The main steering gear control is located in the chart house. A hand-steering wheel is located on the after deck house.

The steam steering gear is capable of putting the rudder from the hard-over to hard-over in twenty seconds when the vessel is steaming at full speed.

ANCHOR ENGINE.

The anchor engine is a two-cylinder, differential valve type, designed to operate at 125 pounds pressure. It is located on the main deck forward of the forward deck house.

DRAINAGE SYSTEM.

For draining the boiler room, engine room and storerooms immediately forward of the boiler room, a 3-inch galvanized pipe is led through these compartments and connected with the fire and bilge pumps in the engine room. Branch lines are taken from the drain piping and led to the lowest point of the compartments and fitted with strainers and stop-

check valves. Suctions are located in forward and after ends of the engine room and fire room. The compartment forward of the fire room has one suction. There is a six-inch suction and piping in the engine room to the circulating pump, which can be used in freeing the engine room of large quantities of water. There is a 2-inch steam ejector in the fire room with a suction in the bilge, and one in the crew's quarters aft with a suction on the platform deck. Other compartments are fitted with stand pipes with connections on the platform deck for the standard Navy handy-billy pump.

FIRE MAIN.

The fire main consists of 2½-inch galvanized piping which runs directly under the main deck from frame No. 31 to No. 95. It connects to two 7-inch × 7-inch × 12-inch vertical, double-acting, simplex Worthington Blake-Knowles fire and bilge pumps located in the engine room. Cut-out valves divide the system into a forward and an after section. Risers with connections lead from the fire main so that every point in the vessel may be reached by a 50-foot length of hose. Water for the flushing system is taken off the fire main.

GALLEY.

The galley installation consists of one two-section oil-burning range, fitted to use either air or steam for atomization, two steam aluminum kettles of twenty gallons each, one coffee urn of twenty gallons capacity, a dresser with sink and steam-heating pipe, and a fuel oil tank with a hand-fuel oil supply pump taking a suction from the fuel oil manifold in the forward part of the fire room.

WEIGHTS.

Group No.	*Description.*	*Weights.*
1	Main turbine, reduction gear, Kingsbury thrust and exhaust connection	21,394
2	Shafting, including casings, sleeves and coupling bolts	5,954
3	Bearings, including bulkheads, stuffing boxes and fairwaters	1,294

6	Main condenser	16,270
7	Main circulating pump with turbine	2,050
	Condensate pump with motor and air ejectors	1,021
8	Propeller	1,550
9–10	Boilers and boiler fittings complete, including burners	50,488
11	Smoke pipes	2,655
	Uptakes	4,490
12	Steam and exhaust pipes and valves	12,504
	Condensate piping	7,329
13	Suction, discharge pipes and valves	7,848
14	Asbestos pipe covering and clothing	5,750
	Sheet metal jacketing	1,500
15	Flooring and grating	5,275
16	Auxiliaries—Main feed pumps	1,900
	Fire and bilge pumps	1,719
	Blowers	2,388
17	Fittings and gear—Feed and filter tank	1,501
	Waste, oil and other small tanks	2,512
	Lifting gear	250
	Gear for operating valves	130
	Gauges, telegraphs, etc	560
	Feed water heater	1,596
	Miscellaneous gear	250
18	Water feed tank	2,417
	condenser	3,005
	feed heater	74
	boilers	8,549
	pumps and pipes	2,175
19	Tools, stores and spare parts (not installed)	
20	Miscellaneous machinery—Air compressor and tanks and piping	1,828
	Distilling plant—evaporator	1,842
	Distillers	2,300
	Heater	625
	Pumps	1,232
	Tanks, piping, water, etc	4,334
	Heating	1,829
	Refrigerating plant, including ice-box	2,250
	Fuel oil pumps, piping, heaters, etc	7,209
	Lubricating oil pumps, piping, coolers, etc	4,418
	Auxiliary condenser and pump	2,952
	Standby pump	188¼
	Fuel oil suction piping to aft tank	273
	Piping, lagging and fitting on auxiliary condenser lines	1,229
	Total, pounds	208,907¼

ELECTRICAL INSTALLATION.

There are two General Electric 10 kilowatt turbo generators, direct connected, 125 volts, 4,000 revolutions per minute. The turbines are designed to operate on full boiler pressure and 10 pounds back pressure.

The main switchboard, distribution panel, and battery charging panel are combined and are located near the generating sets in the engine room. There are three lighting circuits: general, battle and auxiliary. The wiring is Navy Standard.

The auxiliary lighting circuit is arranged to cut in automatically when the quick-closing stop valves are thrown. This circuit is energized by two sets of storage batteries, 32 volts, 210 ampère hour capacity. The auxiliary lights are distributed in parts of the ship where light is essential in case of emergency.

There are two ⅛-horsepower, 32-volt, ventilating sets for forced ventilation. They are operated normally on the 125-volt circuit through a series resistance, which is automatically cut out by relays when it is desired to run on auxiliary circuit.

The radio installation consists of a 1 kilowatt Navy Standard telegraph set and a radio telephone set. Both sets can be controlled from the pilot house. The radio telegraph motor generator is located in the engine room. The telephone is energized from the battery circuit.

In addition to the counter gear, there is an electric tachometer, mounted alongside the gauge board, which indicates the number of revolutions the propeller shaft is actually making and the direction of rotation. The tachometer magneto is driven from a split gear on the propeller shaft.

Reports which have been received concerning the operations of *Eagles 1, 2* and *3* which have been doing duty in the White Sea state that those boats have been doing despatch duty between Murmansk and Archangel, each cruising about 8,000

miles at a standard speed of 14 knots in waters full of floating ice, that they have satisfactorily carried out target practice at a speed of 17 knots, and that they are satisfactory in all respects.

REMARKS ON THE PROBLEM OF IMPACT.

By N. W. Akimoff, Associate.

The problem of impact contains many elusive points. In an average text book it covers but a few pages of space, and we are started off with the following hazy notions:

1. We know something about the coefficient of restitution.

2. Carnot's theorem is given to us as a firm basis for solution of almost any problem: *kinetic energy, lost in impact, is equal to that, due to lost velocity.* This, as a rule, is not so at all, in ordinary problems of life, although, in itself, from the standpoint of theory, Carnot's theorem is, of course, quite correct.

3. We also retain, too well, as a rule, that in impact something is just twice the value it would have if the force were applied gradually. Whether this *something* is the stress or the deflection, and what the other conditions of the problem are, does not matter; but the ratio itself, *twice,* survives longer, generally, than any other notion acquired in the school; and we are actually apt to hypnotize ourselves into the belief that a weight of 2 pounds if dropped on the floor from the height, say, of 5 feet, will produce the *force of blow* exactly equivalent to a 4-pound weight gently placed on the said floor. Anyone who wishes to be cured from this absurd notion is invited to experiment by placing, and then dropping, the weight on his toes; he will see the point at once.

In general, the idea of *force of blow,* and the rather closely associated one, of *work of blow,* mean *absolutely nothing;* in themselves, such things do not exist. It is like that other commonplace question one is often asked, in reference to a wound music-box spring: *how many horsepower is there in such a spring?* There can positively be no answer to such a

question in itself, unless one knows the rate at which the work will be given out by the spring.

The object of these few pages is to illustrate the above in an elementary manner. Some of the arguments have been

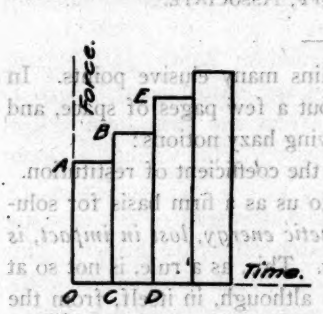


Fig-1.
Momentum as area.

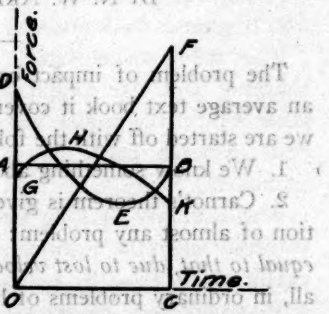


Fig-2.
Various manners in which the same momentum can be produced.

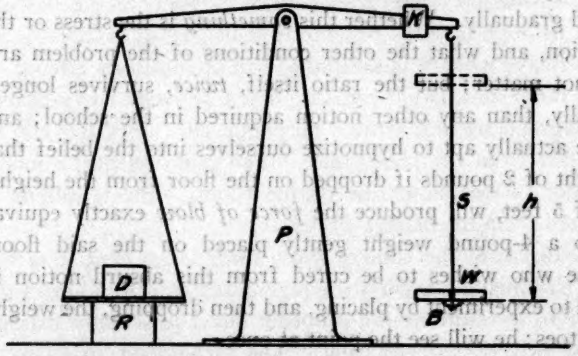


Fig-3.
Prof. Bouasse's experiment.

adapted from Bouasse's "Théories de la Mécanique" (Paris, 1895), while the drop experiments have been performed in writer's Laboratory, by his assistant engineers, Messrs. F. G. Hechler and G. W. Klinger.

It is a well-known theorem of dynamics, that $Ft = mv$, in other words, the product of the force by the time, during which the force acts, is equal to the momentum acquired by the body (or to the increase in momentum, if the body was in motion to begin with).

The graphical illustration of this principle would be as follows: If forces are laid off as ordinates (Fig. 1) and time on the axis of abscissae, it is evident that the area of the rectangle A-C will represent to a certain scale the momentum imparted to the body by the force OA which has acted through the time-interval OC. Similarly, the area of the rectangle BD will represent an increase of momentum, due to the (suddenly increased) force, BC, acting during the time-interval CD, etc. This can, of course, be extended to gradually varying forces, in which case the upper boundary would be a smooth curve. In other words, momentum can always be represented as an area; hence the conclusion, that, being given a certain amount of momentum (a certain area) we are at liberty to imagine a great variety of combinations of force with time, each of which would answer.

For instance, if the momentum, acquired by the body (or lost, for that matter) is expressed as the rectangle OB (Fig. 2), this may have been due:

1. To the constant force OA, acting during OC seconds.
2. To the decreasing force, varying along DEB.
3. To the uniformly increasing force OF.
4. To the variable force GHK,

etc., with the only proviso, that the area OB is maintained.

Furthermore, we do not have to limit ourselves to the same time element OC, but can readily imagine that the same area, OB, can be built up on a very short time interval, which, of course, would bring into play much greater forces. Thus, in order to acquire the velocity of 2,500 feet per second, a shell must be let fall during practically 78 seconds; the same shell, acquiring the muzzle velocity of 2,500 feet per second,

in say, one one-hundredth of a second, must be acted upon by a force 7,760 times the weight of the shell (assuming, for the sake of illustration, the latter force to be constant, which means that the *maximum* force may have been even *greater*). It is quite clear, therefore, that the action of an instantaneous force (which, in reality, mean quick-acting force, and not mathematically instantaneous one) in no manner differs from that of a force acting slowly; only the magnitude of the latter will naturally be high, because the momentum area is to be maintained.

Now, as to the force of blow: the great French philosopher Descartes wrote, as far back as 1640, the following lines in a letter to a friend: "To determine how much weight would be required to equal the force of a hammer blow, is a question of fact, where pure reasoning is of no avail without experiment."

Such an experiment, in the form suggested by Prof. Bouasse, can be performed as follows (Fig. 3): The right scale-pan of the scale P has been removed, and its weight made up by means of the counterweight K; on the string S is suspended a button B and a sliding weight W. The left pan is loaded with the weight D which is heavier than the combined weight of W, S and B, so that the left pan is in contact with the rest R. If there is such a thing as *force of blow*, it is natural to suppose that the weight W, lifted through a certain height *h*, and then dropped on the button B, thus producing the effect of a force, will lift, momentarily, the left pan from the rest. As a matter of fact, we observe nothing of the sort: the height *h*, alone, does not mean anything definite; the effect, the *force of the blow* depends on the elastic properties of the string S. If the height *h* has been found for an inextensible string, it will not produce the desired effect if a yielding string is substituted. In other words, the *momentum itself* tells us absolutely nothing with regard to the force of the blow, unless we prescribe other conditions as well.

In some of the text books the conditions prescribed are such that they *positively cannot* be met with in practice, and a great deal of harm is thereby accomplished to the student's mind. With regard to the action of impact on beams, the following general remarks may be made: if the load W is gradually applied, the deflection will increase gradually, and will reach some final value f; if *the same load* is applied suddenly, the deflection will momentarily reach the value $2f$, and, of course, the corresponding stress will be twice the value due to gradual application; however, the beam will quiet down, after a few vibrations, to the same value f as before, and the stress will be decreased accordingly, *i.e.*, to one-half of the maximum value. All this applies to the sudden application of the weight W, *not of any greater load*, and *not* to dropping of W *from* any height; for instance, let the load W rest in contact with the beam, while its actual weight is taken up by a string; if now the string is cut, we have the case of a sudden application of the load W. To this we might properly refer as dropping from zero-height. If dropped from greater height than zero, the weight will develop much greater stress and cause a greater deflection than the values we just had. How much greater, depends on conditions (see excellent papers, by H. D. Tieman, "Journal Franklin Institute," 1909, also by Armin Elmendorf, *ibid.*, 1916).

As a rule, the following formula, given by St. Venant, gives satisfactory results:

$$f_r = f_s + \sqrt{f_s^2 + f_s \cdot \frac{v^2}{g} \cdot \frac{m}{m+n}},$$

where f_r is the maximum deflection, due to dropping of the weight.

f_s static deflection, corresponding to gradual application of the same load.

v velocity of the load when striking the beam.

m ratio of weight of the load to that of the beam.

n correction coefficient, depending on the method of fastening of the ends of the beam; also on the location of the point at which the beam is struck (St. Venant, "Élasticité des Corps Solides," p. 490, etc.).

This formula, although only approximate, means that both the inertia of the beam and the conditions of the ends have something to do with the value of the greatest deflection (and, therefore, of the greatest stress) due to the impact.

As regards plates, such as floor slabs, armor plates, etc., the matter has not been sufficiently worked out, so far; it is not unreaonable to suppose that the same methods would apply; first, we would assume that the *form* of the deflected surface would not greatly differ from that due to static load; we know enough about deflections of plates to work this out; next, we would determine the potential energy in function of this deflection; finally, the latter would be made equal to the potential energy, due to the following load. The correction, taking into consideration the end conditions and the location of the point of impact, would have to be made along the same lines as was done by St. Venant with regard to beams. All this is difficult, but it most certainly can be done.

The experiments made in writer's Laboratory comprise a set of drop tests, in which the test piece p (see plate) was struck by the sharp point c, forming part of the tup t; the latter was dropped from various heights, as given in the appended tables. The test piece was resting on the stand a, to which we shall refer as anvil, and was made of the following material: hard rubber (valve disk), lead and hard babbitt. In some of the tests the anvil was firmly secured to a heavy cast iron foundation plate; in others, the anvil was placed upon various shock-absorbing substances, such as soft rubber; insulating cork; three soft springs; the characteristics are given below. The "sharp point" C was the apex of a cone, of the following angles: 30 degrees, 45 degrees and 60 degrees. The

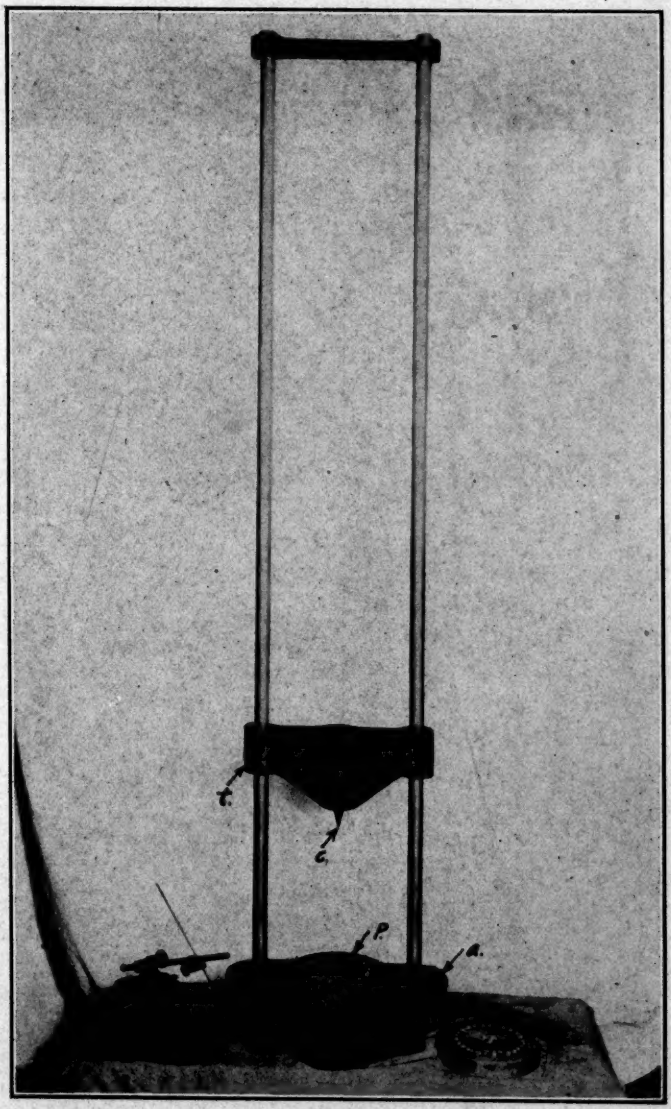


PLATE I.

Note: The figures given in this table represent "Depth of Holes" in inches.

"Sharp Point" Apex of Cone	Anvil mounted on various materials	HEIGHT OF DROP (INCHES) 12 Test Pieces			24 Test Pieces			36 Test Pieces		
		Lead	Babbitt	Hard Rubber	Lead	Babbitt	Hard Rubber	Lead	Babbitt	Hard Rubber
90"	Cast iron	.41	.28	.25	.53	.34	.33	.62	.44	.39
	Cork	.34	.23	.19	.48	.33	.25	.59	.43	.36
	Rubber	.29	.19	.17	.40	.27	.23	.53	.37	.31
	Springs	.28	.15	.15	.37	.26	.19	.50	.35	.28
45"	Cast Iron	.30	.22	.18	.41	.32	.27	.49	.34	.33
	Cork	.28	.19	.16	.40	.25	.23	.48	.31	.28
	Rubber	.25	.16	.13	.36	.24	.20	.44	.30	.25
	Springs	.24	.12	.08	.31	.23	.18	.41	.28	.22
60"	Cast Iron	.27	.18	.16	.36	.23	.20	.43	.27	.25
	Cork	.25	.16	.14	.33	.19	.18	.41	.25	.23
	Rubber	.22	.14	.10	.28	.18	.17	.38	.24	.20
	Springs	.19	.09	.06	.25	.17	.14	.34	.23	.19

ratio of the moving weight to that of the anvil (including the test piece) was maintained constant.

These tests are meant to be a further illustration of the same fact, *i.e.*, that, in itself, there is no such thing as *force of blow* (or *work of blow*); the apparatus shown in the plate, home-made as it is, proved to be satisfactory for this purpose. Of course, it was not designed with the object of competing with Mr. Amsler's well-known drop machine, so that the data, given in the appended table, are of qualitative interest only.

TORSIONAL VIBRATIONS OF IRREGULAR SHAFTS.

By Frank M. Lewis.

ASSISTANT PROFESSOR OF MATHEMATICS, WEBB ACADEMY, NEW YORK.

Engineers have for some years recognized that, under favorable conditions serious torsional vibrations may occur in the shafting of ships. The most important facts that have been ascertained are briefly as follows:

The turning moment of the engine is subject to periodic variations. This period is some multiple of the revolutions, depending upon the type of engine and number of cylinders. In case of doubt as to the period of the principal variations, the turning moment diagram may be analyzed by means of Fourier's Series. * * * See "Synchronous Torsional Vibrations of Shafting," by Hermann Frahm, Journal of the American Society of Naval Engineers, Vol. XIV.

Every shaft has one or more natural periods of torsional vibration. This period is for the case when the shaft is free at both ends.

When the natural period of vibration of the shaft approaches the period of variation of the turning moment of the engine, the vibrations of the shaft tend to increase in magnitude, the stresses in the shaft due to these vibrations increasing until fracture may ultimately occur. Vibrations may also be caused by irregularity of the turning moment of the propeller.

In the design of shafts we should therefore avoid this synchronism and be certain before the engine is built and trouble develops that these vibrations will not occur.

To do this it is necessary to have a method for readily calculating the natural period of torsional vibration of any shaft.

The methods of calculation hitherto proposed are applicable only to a very simple case, namely, a long length of shaft to which is fastened at either end a heavy rotating mass, propeller, engine, turbine, or something similar. Turning to Fig. 4, the shafting of U. S. submarines *S-4* to *S-13*, it is seen that we here have a shaft that can not be considered as even approximating to this simple case. These methods will moreover determine only the period of first order vibrations, or those having one node. As will be seen later, those of secondary order, or two nodes, may be just as important.

We demonstrate in this article a simple graphical method for determining the natural period of torsional vibration of any shaft, no matter in how complicated a manner the distribution of the rotating masses or the stiffness of the shaft itself may vary from end to end. It will moreover determine with equal ease the period of the vibrations of secondary or even higher order, if that is required.

We will discuss first the three general properties of a shaft that determine its period. They are:

The inertia of the rotating masses.

The stiffness.

The length.

The inertia of the rotating masses is measured by the mass polar moment of inertia. The amount of this mass polar moment of inertia per unit of length will be denoted by J_w.

The stiffness of the shaft is measured by the product GJ_s where J_s is the moment of inertia of that portion of the shaft that takes torsion and G is the coefficient of elasticity for shearing. For a section of simple shaft to which nothing is attached it follows that $J_w = wJ_s$, w being the unit weight of the shaft material. If on the shaft there is fastened an armature or other mass so that J_w does not equal wJ_s, that shaft is said to be a loaded shaft.

The use of the product GJ_s as a measure of stiffness follows directly from the formula for the deflection of shafts.

$$\theta = \frac{M l}{G J_s} \qquad (1)$$

M being twisting moment.

l length of shaft.

θ torsional deflection in radians.

That is, torsional deflection under a given twisting moment and length is inversely proportional to GJ_s.

It may happen, as in way of the cranks of an engine, that GJ_s has no direct meaning. If we can determine the torsional deflection of such a shaft under any given twisting moment we can determine by formula (1) an equivalent GJ_s.

We will define a step variable as being a function which remains constant for a certain range of the independent variable and then jumps to a new constant value for the next consecutive range, and so on for its entire length. The graph of some such function might be as in Figure I.

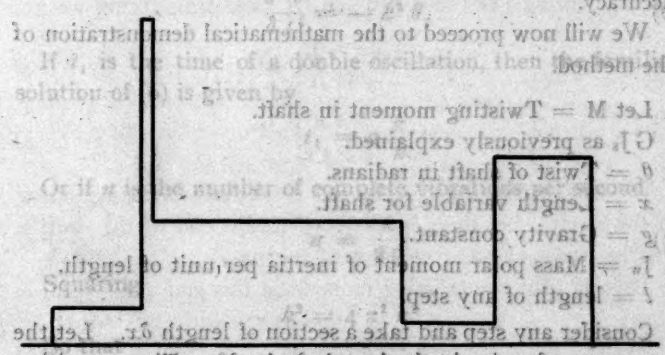


FIG I. GRAPH OF STEP VARIABLE

By choosing suitable breaking points every shaft may be considered as being a step variable.

Over each section or step both J_s and GJ_s are considered as being constant. Unless both the shaft and its load are of uniform dimensions throughout the step, they will, of course

not be constant. In this case equivalent constant values can be calculated.

The equivalent J_w is equal to the total mass moment of inertia of the step divided by the length of the step.

The equivalent GJ_s for the step is determined by finding the total torsional deflection under a given twisting moment. Then from formula (1),

$$GJ_s = \frac{M l}{\theta}$$

will be the equivalent value. The reader will note the step-page used for the shaft in Fig. 4.

The calculator must use his judgment in choosing the breaking points, so that the equivalent shaft thus obtained will closely approximate to the actual one.

In general the use of steps of too short a length will involve a great deal more labor without any compensating gain in accuracy.

We will now proceed to the mathematical demonstration of the method.

Let M = Twisting moment in shaft.
GJ_s as previously explained.
θ = Twist of shaft in radians.
x = Length variable for shaft.
g = Gravity constant.
J_w = Mass polar moment of inertia per unit of length.
l = length of any step.

Consider any step and take a section of length δx. Let the increment of twist in the length δx be $\delta\theta$. Then rewriting equation (1) we get

$$M = GJ_s \frac{\delta\theta}{\delta x} \qquad (1a)$$

Differentiating this we get

$$\delta M = GJ_s \frac{\delta^2\theta}{\delta x^2}\,\delta x \qquad (2)$$

Let the acceleration of this small section be $\frac{\partial^2 \theta}{\partial t^2}$

Then the increment of twisting moment in the length ∂x due to the inertia forces will be

$$\partial M = \frac{J_w}{g} \cdot \frac{\partial^2 \theta}{\partial t^2} \partial x \tag{3}$$

Equating (2) and (3) we get

$$G J_s \frac{\partial^2 \theta}{\partial x^2} = \frac{J_w}{g} \cdot \frac{\partial^2 \theta}{\partial t^2} \tag{4}$$

or

$$\frac{\partial^2 \theta}{\partial x^2} = \frac{J_w}{g G J_s} \cdot \frac{\partial^2 \theta}{\partial t^2} \tag{5}$$

Equation (5) is in the form of the general equation for all harmonic motion. We are only interested however in the simple harmonic motions that can take place.

For simple harmonic motion the general equation is

$$\frac{\partial^2 \theta}{\partial t^2} = -k^2 \theta \tag{6}$$

If t_1 is the time of a double oscillation, then the familiar solution of (6) is given by

$$t_1 = 2 \frac{\pi}{k}$$

Or if n is the number of complete vibrations per second

$$n = \frac{k}{2\pi} \tag{7}$$

Squaring

$$k^2 = 4 \pi^2 n^2$$

So that

$$\frac{\partial^2 \theta}{\partial t^2} = -4 \pi^2 n^2 \theta \tag{8}$$

Substituting (8) in (5) we get

$$\frac{\partial^2 \theta}{\partial x^2} = -\frac{4 \pi^2 J_w}{g G J_s} n^2 \theta \tag{9}$$

The general solution of (9) is given by the equation

$$\theta = a \cos (\beta x + \gamma) \tag{10}$$

From which by differentiation

$$\frac{\delta \theta}{\delta x} = \alpha \beta \sin (\beta x + \gamma) \tag{11}$$

$$\frac{\delta^2 \theta}{\delta x^2} = -\alpha \beta^2 \cos (\beta x + \gamma) \tag{12}$$

α, β and γ are constants. They must be determined for each step so as to satisfy, first the differential equation for that step, and second, the end conditions for that step. It will be noted that n, the vibrations per second, which is the answer we are seeking, enters into equation (9). It will therefore enter into the constants β and γ in every step. The value of n, which enables us to satisfy every differential equation and every end condition throughout the entire length of shafting, will be the solution of the problem which we are seeking. To simplify the explanation we will assume for the present that this value of n is known.

Substituting (10) and (12) in (9) and simplifying we get

$$\beta^2 = \frac{4 \pi^2}{g} \cdot \frac{J_w}{G J_s} n^2 \tag{14}$$

Or

$$\beta = \frac{2 \pi n \sqrt{J_w}}{\sqrt{g G J_s}} \tag{15}$$

We will investigate first the end conditions at the beginning of the shaft. Let there be fastened to this end of the shaft a rotating mass of polar moment of inertia J_1.

Then if M is the twisting moment at this end

$$M = \frac{J_1}{g} \cdot \frac{\delta^2 \theta}{\delta t^2} \tag{17}$$

Also

$$M = G J_s \cdot \frac{\delta \theta}{\delta x} \tag{1a}$$

Equating

$$\frac{J_1}{g} \cdot \frac{\delta^2 \theta}{\delta t^2} = G J_s \cdot \frac{\delta \theta}{\delta x} \tag{18}$$

From (8)

$$\frac{\delta^2 \theta}{\delta t^2} = -4 \pi^2 n^2 \theta$$

From (10) $\theta = a \cos(\beta x + \gamma)$

From (11) $\frac{\delta \theta}{\delta x} = -a \beta \sin(\beta x + \gamma)$

Substituting these values in (18) we get

$$\cos(\beta x + \gamma) \frac{4 \pi^2 n^2 J_1}{g G J_s} = \beta \sin(\beta x + \gamma) \quad (19)$$

Since we are considering the beginning of the shaft, $x = 0$ and (19) reduces to

$$\tan \gamma = \frac{2 \pi n J_1}{\sqrt{g G J_s J_w}} \quad (20)$$

This is the condition to be satisfied at the beginning of the shaft.

If nothing is attached to the shaft at this point $\tan \gamma$ reduces to zero.

We have now determined the constants β and γ for the first step in terms of n and the known properties of the shaft. a can have any value for this step. It is a measure of the amplitude of vibration at the beginning of the shaft.

We will now investigate the end conditions for the second step. We will use the subscript 1 to denote the first step, 2 the second step, etc.

From (15) $$\beta_2 = \frac{2 \pi n \sqrt{J_{w_2}}}{\sqrt{g G_2 J_{s_2}}} \quad (21)$$

At the end of the first step

$$M_1 = G_1 J_{s_1} \frac{\delta \theta}{\delta x} \quad (22)$$

Substituting the values for $\frac{\delta \theta}{\delta x}$ and β_2 given by (11) and (21) in (22) we get

$$M_1 = -a_1 \frac{2 \pi n \sqrt{G_1 J_{s_1} J_{w_1}}}{\sqrt{g}} \sin(\beta_1 l_1 + \gamma_1) \quad (23)$$

Likewise at the beginning of the second step, since for this point $x = 0$

$$M_2 = -\frac{a_2\, 2\pi n \sqrt{G_2 J_{s_2} J_{w_2}}}{\sqrt{g}} \sin \gamma_2 \qquad (24)$$

l_1 being the length of the first step.

The turning moments on each side of the break are equal. Therefore equating M_1 and M_2 and clearing we get

$$a_1 \sqrt{G_1 J_{s_1} J_{w_1}} \sin(\beta_1 l_1 + \gamma_1) = a_2 \sqrt{G_2 J_{s_2} J_{w_2}} \sin \gamma_2 \qquad (25)$$

Also at the break the deflections must be equal, or

$$a_1 \cos(\beta_1 l_1 + \gamma_1) = a_2 \cos \gamma_2 \qquad (26)$$

We will now show how these equations and conditions can be expressed graphically. Turning to Fig. 2 we lay off a distance OA of length equal to unity. Lay off a perpendicular AC equal to $\tan \gamma$ as given by equation (20). Then $\angle$ COA $= \gamma$ and OC $= a$.

Substituting the value of β given by (15) in (10) we get

$$\theta = a_1 \cos\left\{\frac{2\pi n \sqrt{J_{w_1}}}{\sqrt{g\, G_1 J_{s_1}}} x + \gamma_1\right\} \qquad (27)$$

The angles are in radians. It will be convenient to put them in degrees. In degrees

$$\theta = a_1 \cos\left\{\frac{360\, n \sqrt{J_{w_1}}}{\sqrt{g\, G_1 J_{s_1}}} x + \gamma_1\right\} \qquad (28)$$

When $x = l_1$

$$\theta = a_1 \cos\left\{\frac{360\, n_1 \sqrt{J_{w_1}}}{\sqrt{g\, G_1 J_{s_1}}} l_1 + \gamma_1\right\} \qquad (29)$$

The expression $\dfrac{360 \sqrt{J_w}\, l}{\sqrt{g\, G\, J_s}}$ should be calculated for

each step. We will call it the angular function and denote it by the symbol ϕ

$$\phi = \frac{360\ \sqrt{J_w}\ l}{\sqrt{g}\ G\ J_s} \qquad (30)$$

l being the length of the particular step with which we are working.

To satisfy equation (29) graphically we then lay off from OC an angle in degrees equal to $\Phi_1 n$ and draw the arc DC.

Erect a perpendicular DE. Then if OA represents the deflection at the beginning of the shaft, OE will represent the deflection at the end of the first step.

Let DE be denoted by y_1. Lay off a distance $EF = y_1^1$ so that

$$y_1^1 = y_1 \frac{\sqrt{G_1\ J_{s1}\ J_{w1}}}{\sqrt{G_2\ J_{s2}\ J_{w2}}} \qquad (31)$$

It is now seen that $OF = a_2$ and $\angle\ FOA = \gamma_2$ for equation (26) is thus satisfied and since

$$a_1 \sin(\beta_1 l_1 + \gamma_1) = y_1$$

and

$$a_2 \sin \gamma_2 = y_1^1$$

we have satisfied equation (25).

Lay off now from OF an angle $\Phi_2 n$ and proceed as for the second step until the end of the shaft is reached, say on the line OJ or OJ^1. At this end of the shaft suppose there to be attached a rotating mass of moment of inertia J_2. Then if M_e is the turning moment due to the acceleration of this rotating mass

$$M_e = -\frac{J_2}{g}\frac{\partial^2 \theta}{\partial t^2} \qquad (32)$$

And

$$M_e = G_e\ J_{se}\ \frac{\partial \theta}{\partial x} \qquad (33)$$

Equating (32) and (33) and proceeding in the same manner as by which we derived equation (19) we get

$$\tan\ (\beta_e\, l_e + \gamma_e) = -\ \frac{2\,\pi\,n\,J_z}{V\,g\,G_e\,J_{se}\,J_{we}} \qquad (34)$$

The e subscripts denoting the end or last step. Equation (34) will be satisfied if

$$\beta_e\, l_e + \gamma_e = \angle\ AOJ + m\,\pi$$

m being o or any integer and $\angle\ AOJ < \pi$

and $$\tan \angle\ A\,O\,J = -\ \frac{2\,\pi\,n\,J_z}{V\,g\,G_e\,J_{se}\,J_{we}}$$

If nothing is attached to this end of the shaft $\beta_e\, l_e + \gamma_e$ will equal $\pi + m\,\pi$

Now for each value of the integer m some value of n will satisfy all the equations and end conditions throughout the shaft and will therefore be one of the natural periods of vibration of the shaft. When $m = 0$, n will be the period of the vibrations of primary order, for we see from the polar diagram that θ will pass through the 0 value once, that is, there will be one node. When $m = I$ we see that θ will pass through the 0 value twice and we will have vibrations of secondary order.

We wish to find, then, the values of n by which the construction in the polar diagram will cover a multiple of 180 degrees. If we are solving for primary vibrations we wish to cover 180 degrees; if for secondary 360; third order, 540 degrees, etc.

The value of n which will do this is determined by trial and error.

We will now review the actual procedure of calculation for easy reference.

(1) The shaft is divided into steps of suitable length.

(2) For each of these steps an equivalent J_e and GJ_e is calculated. In other words, we have substituted for the actual

shaft an equivalent shaft made up of steps of uniform size and loading. Note that masses attached to the extreme ends of the shaft are not considered as belonging to any step.

(3) Calculate for each step the angular function given by equation (30)

$$\phi = \frac{360\sqrt{J_w}\,l}{\sqrt{g\,G\,J_s}}$$

And the break function given by equation (31)

$$= \sqrt{G\,J_s\,J_w}$$

(4) If to the ends of the shaft rotating masses are fastened calculate for each end the tangent function given by equation (20) or (34)

$$\tan\gamma_1 \text{ or } \tan(\beta_e\,l_e + \gamma_e) = \frac{2\pi n J_1}{\sqrt{g\,G\,J_s\,J_w}}$$

(5) Take one end of the shaft as the beginning and the other as the end and calculate for each break the ratio of the break functions, marking them on the diagram of the shaft for convenience.

That is find

$$\frac{y_1^1}{y_1},\ \frac{y_2^1}{y_2},\ \frac{y_3^1}{y_3},\ \text{etc.}$$

(6) Assume now some value for n, and starting at the beginning of the shaft lay off in the diagram OA equal to unity and AC equal to $\tan\gamma_1$.

From OC lay off an angle $DOC = \phi_1 n$, multiply DE in the ratio $\frac{y_1^1}{y_1}$, lay off $\angle FOG = \phi_2 n$, and proceed thus until the end of the shaft is reached.

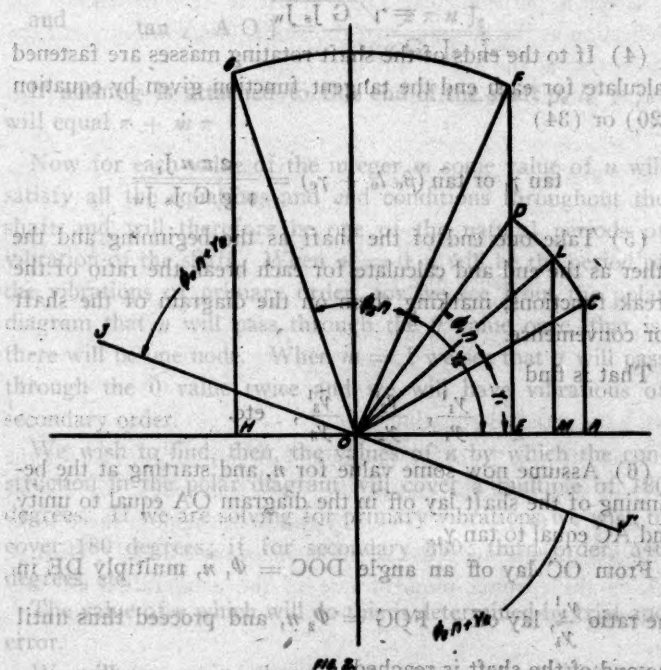
Lay off finally the angle whose tangent has been calculated for the rotating mass at the end of the shaft.

The total angle we have now passed over will be a measure of the closeness of the first assumption for n from which a

second approximation can be made, and thus on to the required degree of closeness. Or we might plot a curve of total angle and n and find for what values of n this curve cuts the 180 or 360 degree line, whichever is required.

Once the functions have been calculated for each step the approximate diagrams can be constructed very rapidly.

It is apparent at once that a diagram that fulfills the necessary conditions, that is, that covers a multiple of 180 degrees,



is a diagram of relative angular shaft deflections due to torsional vibration. In Fig. 2, for instance, OA is the deflection at the beginning of the shaft, OE the deflection at the end of the first step, OH at the end of the second step, etc. The

deflection at any point within the step is found by dividing the angle for the step, as COD, in the same ratio as the point divides the length of the step. To find the deflection at the middle of the first step, for instance, draw OL bisecting angle DOC. Then OM is the deflection.

In this manner the position of the nodes is instantly determined, a node being a point of zero deflection is the point on any step where its arc cuts the vertical through the origin.

We will illustrate the calculations for a simple case.

To one end of a shaft 122 feet long and 10 inches diameter, is fastened a propeller of J equal 3,550 in foot pound units, and to the other end a gear of J equal 12,500. Determine the period.

As there is but one step it will not be necessary to construct a polar diagram.

For the shaft,

$$J_s = \frac{\pi}{32} \times \frac{10^4}{12^4} = .0474$$

At the gear end

$$\tan \gamma_1 = \frac{2\pi \times 12{,}500 \times n}{\sqrt{32.16 \times 12{,}000{,}000 \times 144 \times 490 \times .0474}} = .318n$$

At the propeller end

$$\tan \gamma_2 = \frac{2\pi \times 3550 \times n}{\sqrt{32.16 \times 12{,}000{,}000 \times 144 \times 490 \times .0474}} = .0904n.$$

And

$$\varphi n = \frac{360 \times \sqrt{490} \times 122 \times n}{\sqrt{32.16 \times 12{,}000{,}000 \times 144}} = 4.11\, n$$

We determine then the values of γ_1, γ_2 and Φn for various values of n and tabulate

n	γ_1	γ_2	Φn	$\gamma_1 + \gamma_2 + \Phi n$
1	17.65	5.4	4.11	27.16
5	57.8	24.3	20.55	102.65
10	72.5	42.1	41.1	155.7
15	78.2	53.7	61.5	193.4
20	81.0	61.0	82.2	224.3
30	84.0	69.8	123.3	277.1
40	85.5	74.5	164.4	324.4
50	86.4	77.5	205.5	369.4

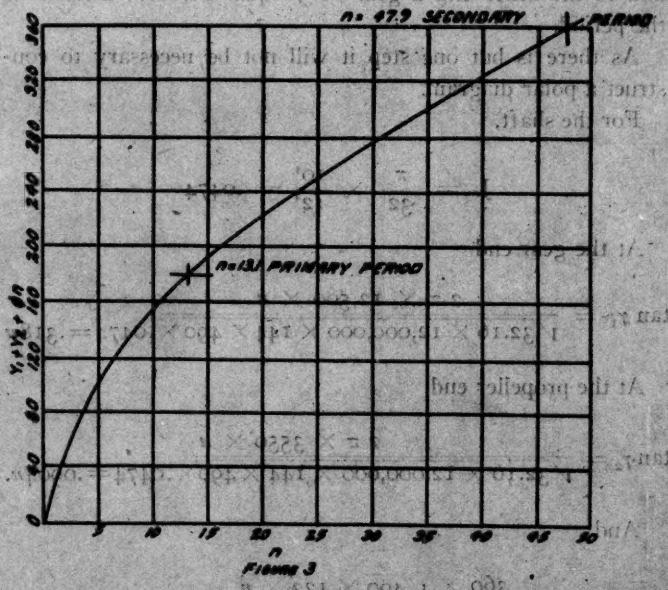
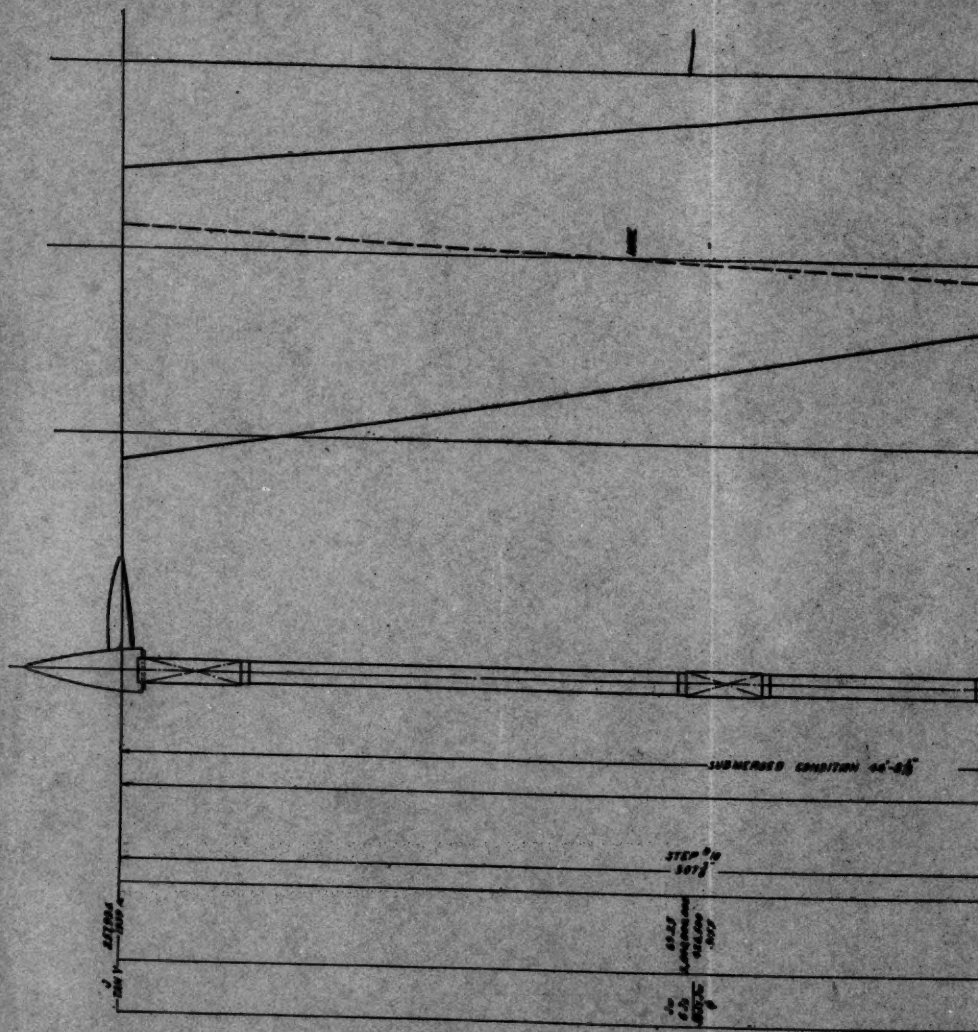
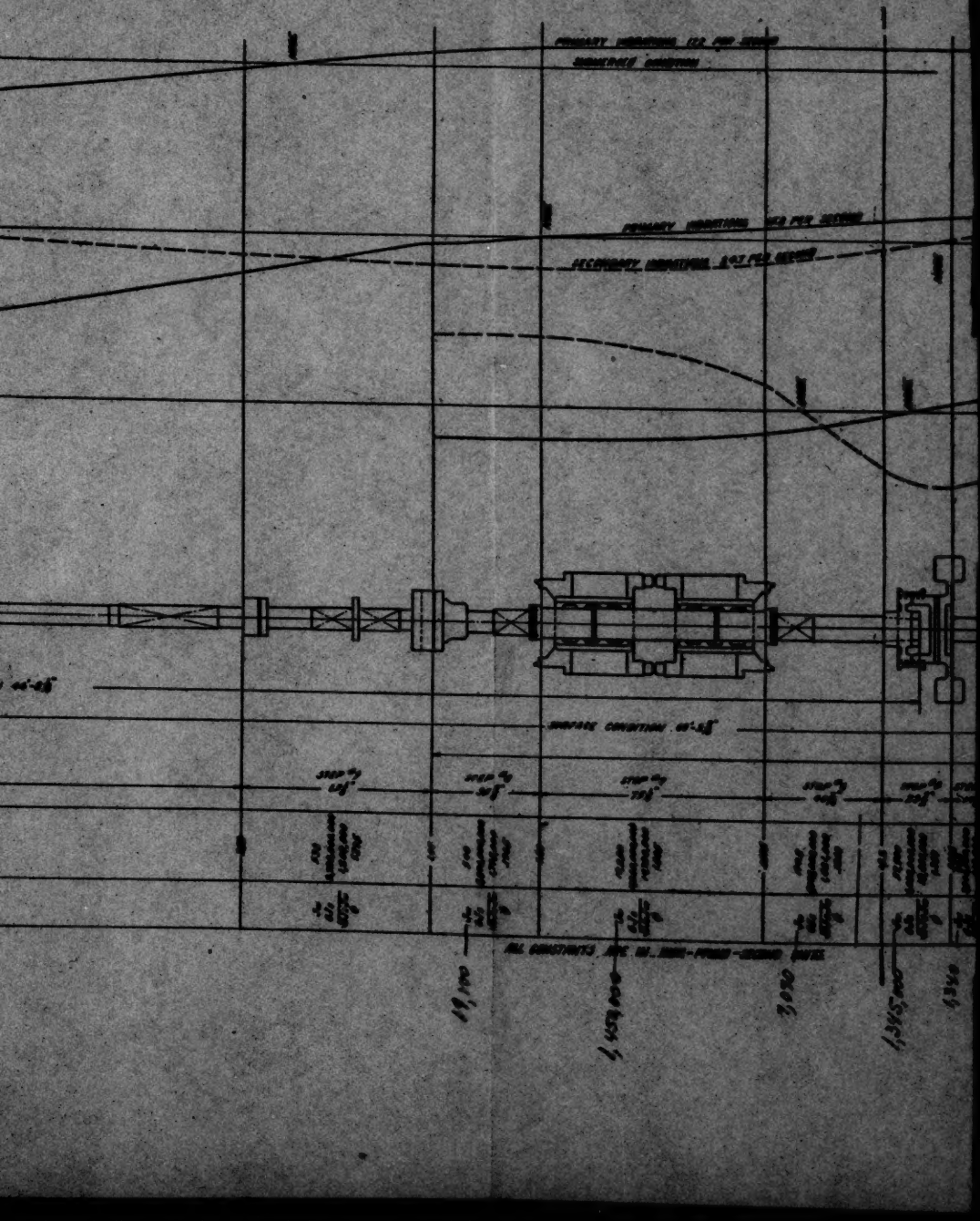
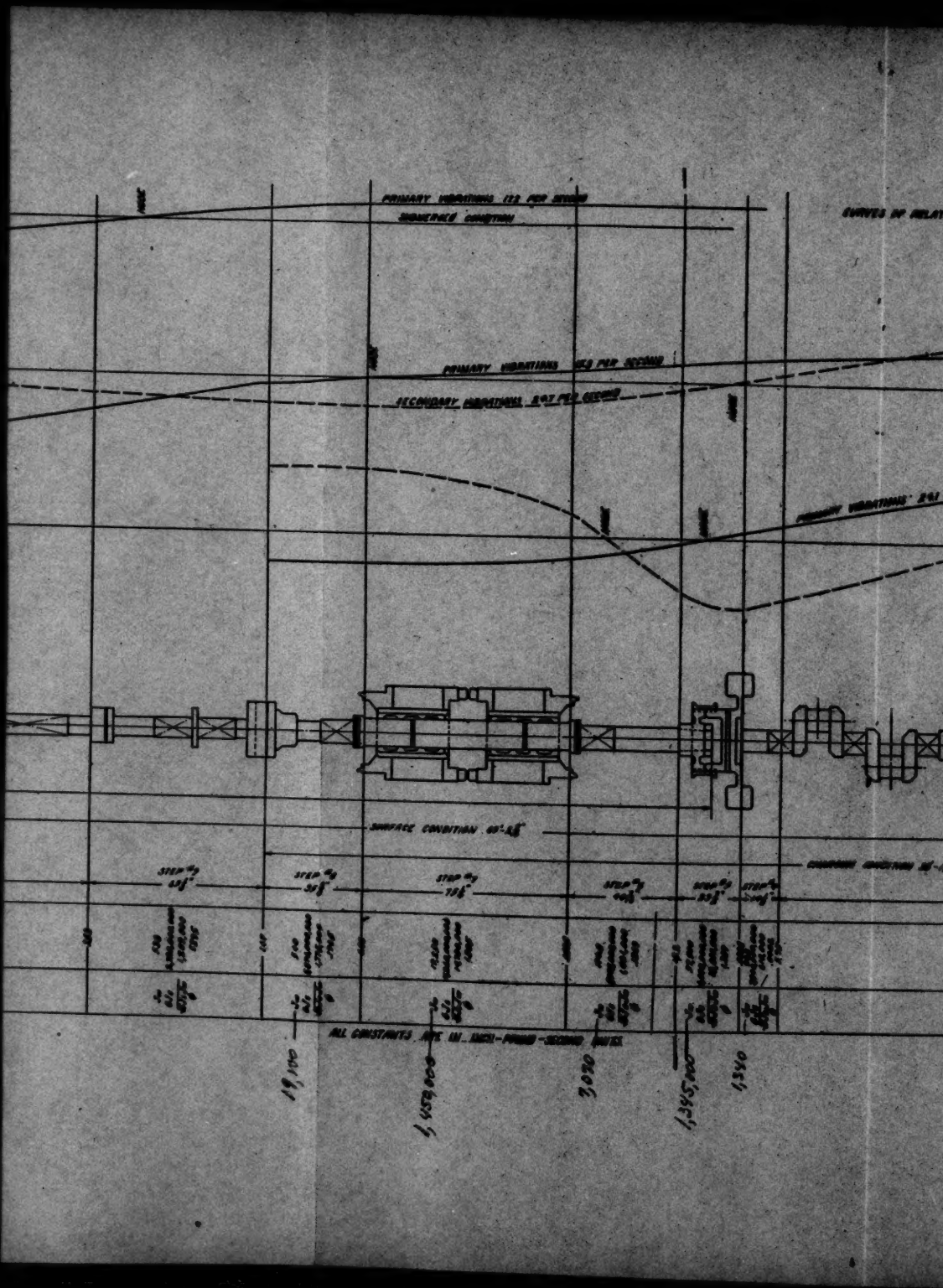


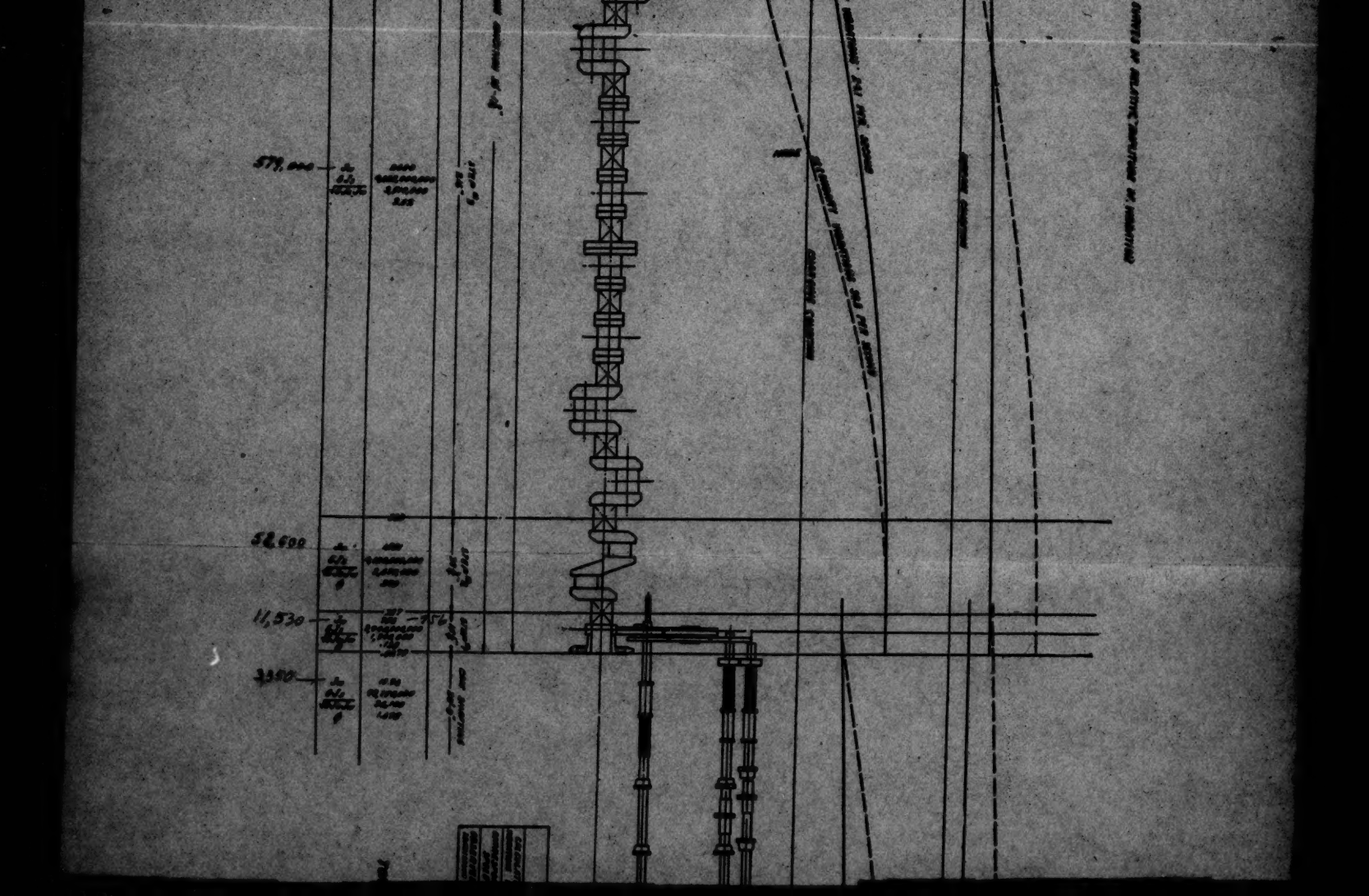
FIGURE 3

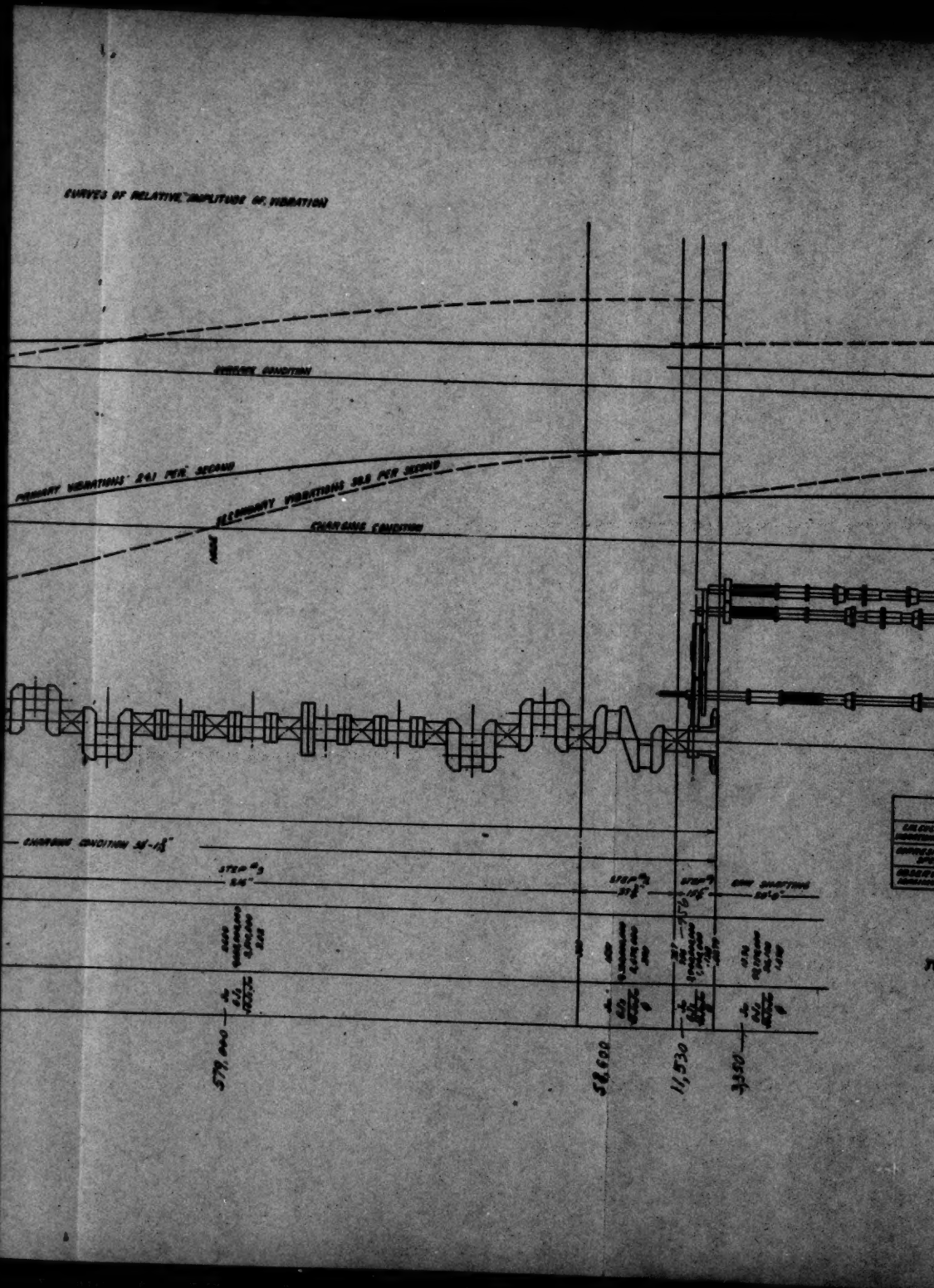
In Fig. 3, $\gamma_1 + \gamma_2 + \Phi n$ is plotted to a base of n. We see that for $\gamma_1 + \gamma_2 + \Phi n = 180$ degrees, $n = 13.1$; and that for $\gamma_1 + \gamma_2 + \Phi n = 360$ degrees, $n = 47.9$. The primary period is therefore 13.1 vibrations per second and the secondary 47.9.

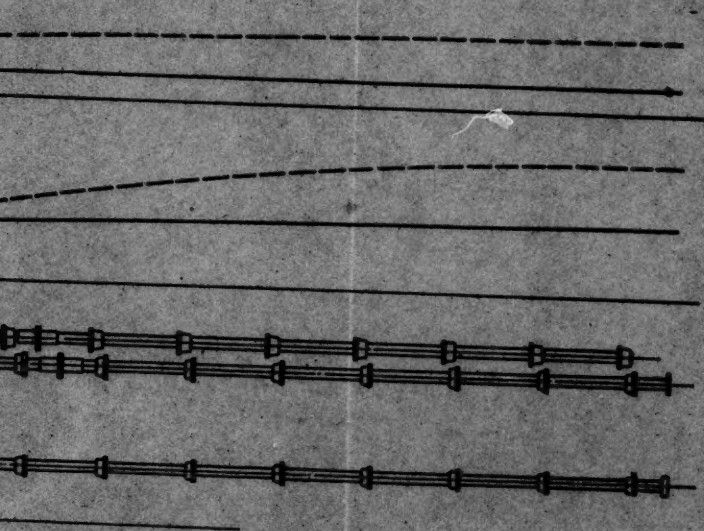

SUBMERGED CONDITION 46'-0"




PRIMARY VIBRATIONS
SURFACE CONDITION
PRIMARY VIBRATIONS
SECONDARY VIBRATIONS
SURFACE CONDITION
ALL CONSTANTS ARE IN INCH-POUND-SECOND UNITS.
19,100
1,458,800
3,020
1,345,700
1,340


579,000
58,600
11,530
3350


CURVES OF RELATIVE AMPLITUDE OF VIBRATION
SURFACE CONDITION
PRIMARY VIBRATIONS 24.1 PER SECOND
SECONDARY VIBRATIONS 48.2 PER SECOND
CHARGING CONDITION
CHARGING CONDITION 36'-1 1/2"
579,000
58,600
11,530
3350



	SHAFT				
	CLUTCH		SURFACE		PROPELLER
	FREQUENCY	REVOLUTIONS	ORDER	REVOLUTIONS	REVOLUTIONS
CALCULATED FIRING FREQUENCY PER ORDER	241	30.8	15.8	247	172
CORRESPONDING CRITICAL SPEED R.P.M.	96	385	237	370	
OBSERVED SPEED OF MINIMUM VIBRATION	350			345	

U.S. SUBMARINES SS-313
TORSIONAL VIBRATION ANALYSIS OF SHAFTING

FIG. 4

To determine the position of the primary node we note that for $n = 13.1$,

$$\gamma_1 = 76.5$$
$$\gamma_2 = 49.8$$
$$\phi n = 53.7$$
$$180.0$$

Therefore the primary node is $\frac{90-76.5}{53.7} \times 122 = 30.65$ feet from the gear end.

Also for $n = 47.9$ $\quad \gamma_1 = 86.1$

$$\gamma_2 = 76.9$$
$$\phi n = 197.0$$
$$360.0$$

Therefore the first secondary node is $\frac{3.9}{197} \times 122 = 2.42$ feet from the gear end.

And the second secondary node is $\frac{183.9}{197} \times 122 = 113.9$ feet from the gear end.

The calculations have been made to determine the period of the shafting of U. S. Submarines *S-4*—*S-13* and are illustrated here. The shafting is shown in figure 4. There are three cases to be investigated. First, the charging condition when the engine is driving the dynamo, the after clutch being disconnected. Second, the surface condition, when the engine drives the propeller. Third, the submerged condition when the motors drive the propeller, the forward clutch being disconnected.

The main shaft was divided into ten steps as shown and for each of these J_w and GJ_θ calculated. In finding J_w in way of the cranks one-half of the reciprocating weight was used, and in finding the moment of inertia of the propeller 25 per

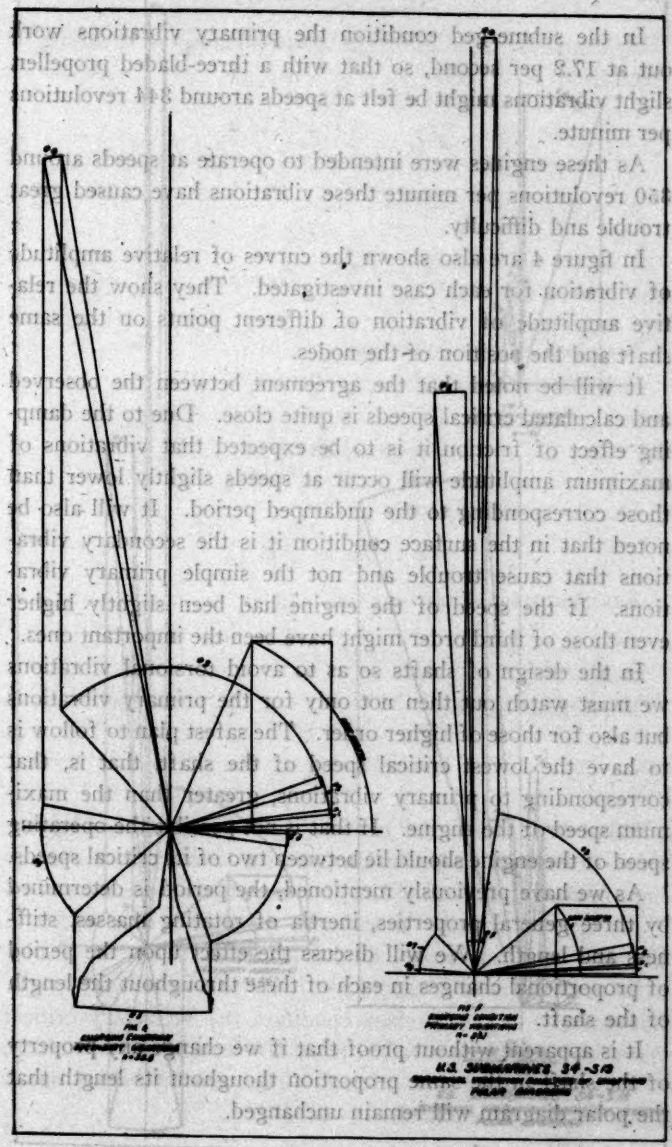
cent was added to the calculated amount to allow for the entrained water.

In calculating GJ_s, G was taken at 11,790,000 * * *. See JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XIV, Page 731. To determine the value of GJ_s in way of the cranks a torsion experiment was made upon the actual engine shaft in its bearings. The value determined by this experiment for GJ_s was 4,620,000,000. The value of GJ_s in way of the crank bearings is 3,940,000,000. The shaft with cranks is therefore 17.3 per cent stiffer than the plain shaft. In way of the clutches the value of GJ_s was decreased somewhat to allow for the effects of play.

On each step are marked the values of J_w, GJ_s, $\sqrt{GJ_s J_w}$ and ϕ and on the breaks the ratios from right to left of $\sqrt{GJ_s J_w}$.

The closing polar diagrams for primary and secondary vibrations are shown in figures 5 to 9. In the charging condition for primary vibrations the period was found to be 24.7. As there are 8 cylinders and therefore 4 impulses per revolution, the critical speed would be $24.7 \times 60/4 = 361$ revolutions per minute. In operation, vibrations of great violence were felt at speeds around 350 revolutions per minute, being most violent at speeds between 340 and 360 revolutions per minute. The period of the secondary vibrations for this condition was 38.8, corresponding to a critical speed of 584 revolutions per minute, which is higher than any at which the engine was run.

In the surface condition the primary period works out at 15.8, corresponding to a critical speed of 237 revolutions per minute. The period of the secondary vibrations is 24.7, corresponding to a critical speed of 370 revolutions per minute. In operation violent vibrations were felt at speeds around 365 revolutions per minute.



In the submerged condition the primary vibrations work out at 17.2 per second, so that with a three-bladed propeller, slight vibrations might be felt at speeds around 344 revolutions per minute.

As these engines were intended to operate at speeds around 350 revolutions per minute these vibrations have caused great trouble and difficulty.

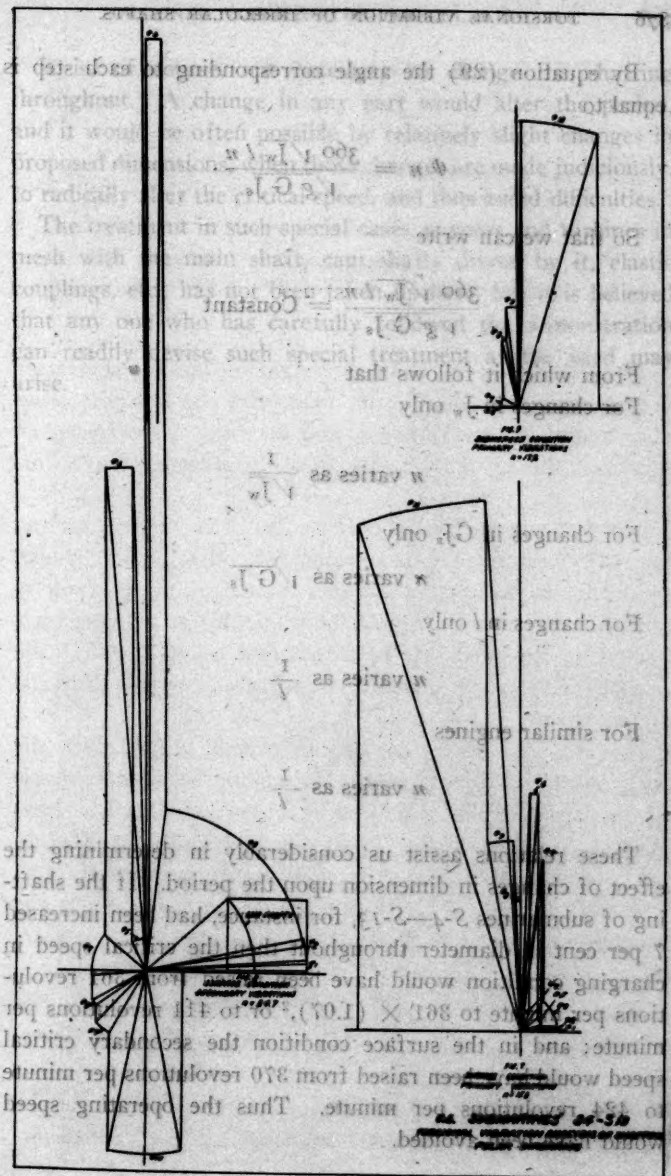
In figure 4 are also shown the curves of relative amplitude of vibration for each case investigated. They show the relative amplitude of vibration of different points on the same shaft and the position of the nodes.

It will be noted that the agreement between the observed and calculated critical speeds is quite close. Due to the damping effect of friction it is to be expected that vibrations of maximum amplitude will occur at speeds slightly lower than those corresponding to the undamped period. It will also be noted that in the surface condition it is the secondary vibrations that cause trouble and not the simple primary vibrations. If the speed of the engine had been slightly higher even those of third order might have been the important ones.

In the design of shafts so as to avoid torsional vibrations we must watch out then not only for the primary vibrations but also for those of higher order. The safest plan to follow is to have the lowest critical speed of the shaft, that is, that corresponding to primary vibrations, greater than the maximum speed of the engine. If that is not possible the operating speed of the engine should lie between two of its critical speeds.

As we have previously mentioned, the period is determined by three general properties, inertia of rotating masses, stiffness and length. We will discuss the effect upon the period of proportional changes in each of these throughout the length of the shaft.

It is apparent without proof that if we change any property of the shaft in the same proportion thoughout its length that the polar diagram will remain unchanged.



By equation (29) the angle corresponding to each step is equal to

$$\phi_n = \frac{360\sqrt{J_w}\, l\, n}{\sqrt{g\, G\, J_s}}.$$

So that we can write

$$\frac{360\sqrt{J_w}\, l\, n}{\sqrt{g\, G\, J_s}} = \text{Constant}$$

From which it follows that

For changes in J_w only

$$n \text{ varies as } \frac{1}{\sqrt{J_w}}$$

For changes in GJ_s only

$$n \text{ varies as } \sqrt{G\, J_s}$$

For changes in l only

$$n \text{ varies as } \frac{1}{l}$$

For similar engines

$$n \text{ varies as } \frac{1}{l}$$

These relations assist us considerably in determining the effect of changes in dimension upon the period. If the shafting of submarines *S-4*—*S-13*, for instance, had been increased 7 per cent in diameter throughout then the critical speed in charging condition would have been raised from 361 revolutions per minute to $361 \times (1.07)^2$ or to 414 revolutions per minute; and in the surface condition the secondary critical speed would have been raised from 370 revolutions per minute to 424 revolutions per minute. Thus the operating speed would have been avoided.

It is, of course, not necessary to change the shafting throughout. A change in any part would alter the period, and it would be often possible by relatively slight changes in proposed dimensions, when those changes are made judiciously, to radically alter the critical speed, and thus avoid difficulties.

The treatment in such special cases as gears and turbines in mesh with the main shaft, cam shafts driven by it, elastic couplings, etc., has not been taken up here, but it is believed that any one who has carefully followed the demonstration can readily devise such special treatment as the need may arise.

U. S. S. *RAMAPO*

(EMERGENCY FLEET CORPORATION OIL TANKER NO. 1655).

DESCRIPTION AND TRIALS.

By HENDERSON B. GREGORY, ASSOCIATE.

The *Ramapo* is the second of eight oil tankers contracted for, under date of October 10, 1918, with the Newport News Shipbuilding & Dry Dock Co. and the Navy Department, on behalf of the U. S. Shipping Board Emergency Fleet Corporation.

Under the terms of the contract the price of the hull and machinery was based upon an estimated cost of $2,200,000.00, plus or minus the cost of authorized changes, the contractors to receive a fixed profit of $220,000.00; this profit to be augmented, in the event of the actual cost being less than the estimated cost or authorized revision thereof, by one-third the amount saved.

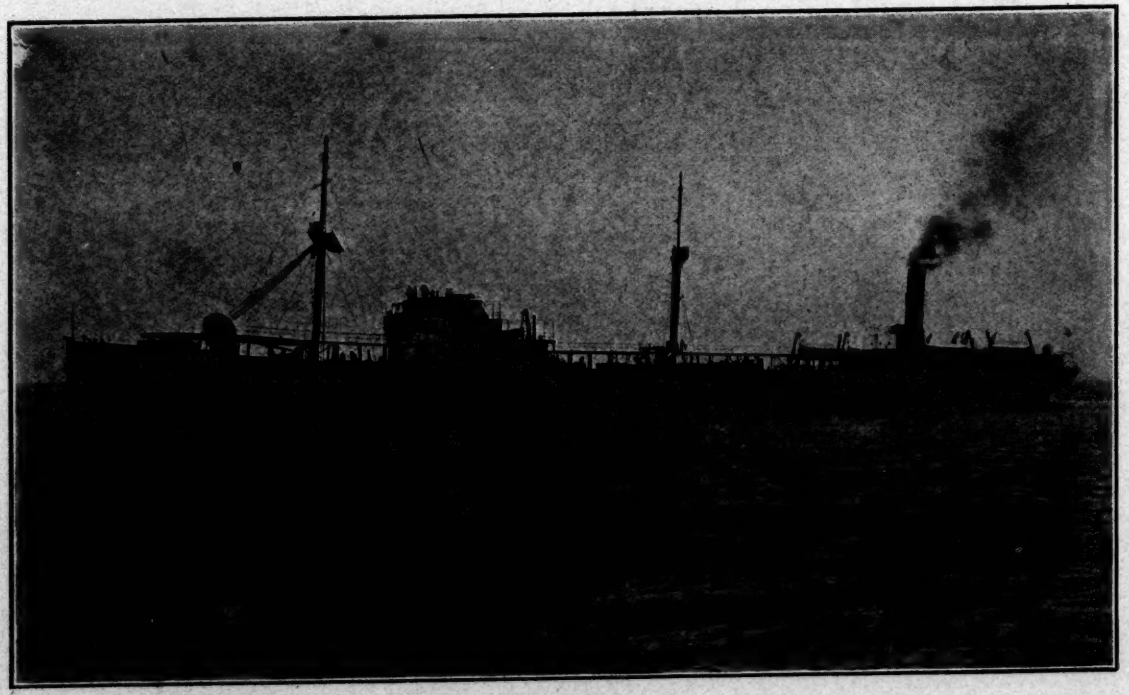
The hull and machinery are in general accord with the Rules of the American Bureau of Shipping for highest classification, and the U. S. Steamboat Inspection Rules for seagoing vessels. The Navy Department also inspected all work.

The keel was laid January 16, 1919, and vessel was launched September 11, 1919. Thirty-four days after the launching the vessel proceeded to sea for her trials.

GENERAL DESCRIPTION OF HULL.

The hull is of the following principal dimensions:

Length over all, feet and inches	477–10
between perpendiculars, feet and inches	463–3
on load-water line, feet and inches	462–10



U. S. S. "RAMAPO."

Beam, extreme, feet and inches	60–3¼
on load-water line, feet and inches	60–0
moulded, feet and inches	60–0
Mean designed draught, feet and inches	26–2
Displacement on designed draught, tons	16,785
per inch on load-water line, tons	57.9
Area of immersed midship section at designed draught, square feet	1,551
of load-water line plane at designed draught, square feet	24,316
Wetted surface at designed draught, square feet	46,000
Coefficient of fineness, block	0.812
midship section	0.988
load-water line	0.873

The vessel is a single screw, steel steamship, with straight stem, elliptical stern and with flat sheer amidships. It is rigged with two masts and one smoke pipe. It is of the shelter-deck type, with complete steel shelter, upper and main decks, except that the upper and main decks are omitted in way of the oil tank expansion trunks and the machinery space. The longitudinal system of framing and scantlings is followed throughout.

The propelling machinery is located in the after end of the hold, and a screen bulkhead separates the engine and boiler rooms. Immediately abaft the machinery compartment is the after peak tank. Forward the boiler room and separated therefrom by a cofferdam, are the fuel oil bunker tanks, of 63,000 cubic feet capacity.

Reserve feed water is carried in tanks under the machinery space, and there is a ballast tank beneath the fuel oil bunkers.

The amidship hold is divided into nine tanks, of about 450,000 cubic feet total capacity, for carrying oil in bulk. The tanks are provided with center-line bulkhead extending to the shelter deck, dividing them into port and starboard compartments, and there are expansion trunks between the main and shelter decks for the full length of the tanks. The wing spaces, between the main and upper decks in way of the main oil tanks, are divided into five compartments, port and starboard, fitted

up as summer oil tanks. The total capacity of these tanks is 43,000 cubic feet. There is a pump room between oil tanks Nos. 5 and 6, and a cofferdam forward of tank No. 1, between tanks Nos. 3 and 4, and between tank No. 9 and the fuel oil bunker tanks.

The following are located in the forward hold: a pump room, cargo space, fuel oil or ballast tanks, chain locks and the fore peak tank.

Aft on the main deck are located the cold storage compartments, fresh water tanks and storage space. Forward are cargo space and stores.

The upper deck aft is fitted as crew's quarters. The sick bay is located on this deck on the starboard side of the engine hatch. Forward are crew's quarters, prison, carpenter shop, lamp room, emergency generator room, paint room and windlass engine.

Aft on the shelter deck is a deck house for the steering gear, with gun platform on top of the deck house. A similar gun platform is provided in the bow over the windlass. No guns are mounted.

The crew's mess is located in a deck house abreast of the engine hatch, port side, on the shelter deck, with spacious galley on center line abaft the engine hatch. The corresponding deck house on the starboard side of the engine hatch contains the general mess pantry, stores, laundry, and engineer's office.

The radio room is on top of the after deck house.

Forward is a double tier deck house containing the officers' quarters. It is raised 42 inches clear of the shelter deck. On top of this deck house are the wheel house, chart room and bridges.

The tops of all deck houses and gun platforms are connected by fore-and-aft bridges.

LIFE BOATS, ETC.

The following life boats and life rafts are carried in skid-beams on top of the after deck houses, except the whale boats, which are carried on the forward deck house:

No.	*Type.*	*Capacity each.*
4	26-ft. Metallic life boats	50 persons
2	24-ft. Whale boats	23 persons
1	24-ft. Motor boat	
1	Racine life raft	24 persons
2	Life buoy rafts	25 persons

All boats are swung in Norton sheath screw davits.

DECK MACHINERY.

Windlass.—A 14-inch × 12-inch, double engine, Hyde anchor windlass, with four gipsy heads, is located on the shelter deck forward; the engine is below on the upper deck. Patent anchors are carried, which are stowed in the hawse pipes.

Deck Winches.—Three deck winches are provided; they are all of the Hyde Windlass Company's design. The bow and midship winches have two 8-inch × 10-inch steam cylinders; the stern winch, which is of greater power, has two 9-inch × 14-inch steam cylinders. All winches have two gipsy heads.

Steering Engine.—The steeering engine and gear are located in a deck house aft on the shelter deck. It is of the Brown Steam Tiller type, as manufactured by the Hyde Windlass Co. The steam cylinders are 9-inch × 12-inch.

Ammunition Hoists.—Four small Lidgerwood ammunition hoists are provided, two for each gun. They are driven by G. E. Co. motors. The ammunition is carried in small magazines in the forward and after holds.

CARGO OIL PIPING.

The cargo oil suction system consists of two 11-inch steel pipe lines, leading aft from the pump room, one on each side of

the center line bulkhead, for draining tanks 6 to 9, inclusive. Four similar lines lead forward, two for tanks 1, 2 and 3, and two for tanks 4 and 5. Each tank connects to its main through a 9-inch angle valve fitted with bell mouth carried down to near the bottom of the tank. A master gate valve is fitted in the suction main at the transverse bulkhead nearest the pump room of each tank served. All valves are operated from the shelter deck. Valves are arranged that the pumps may draw from any tank and discharge to any other tank.

A 16-inch gate valve, operated from the shelter deck, is fitted on the center-line bulkhead between each pair of main cargo tanks.

The summer tanks are drained by 6-inch pipes. Four 6-inch filling pipes, two port and two starboard, terminating on the shelter deck, are provided for these tanks; they connect to the 6-inch drain mains.

Two 11-inch risers are fitted forward, from the suction mains to the shelter deck, for filling tanks 1, 2 and 3, and two similar risers aft for tanks 6 to 9, inclusive. Tanks 4 and 5 are filled through connections on the 11-inch discharge mains at the pump room hatch, port and starboard. Each filling connection terminates at the deck with an 11-inch gate valve and 11-inch × 8-inch × 8-inch Y-casting with flanged ends. Eight-inch × 6-inch flanged reducers are provided to suit 6-inch U. S. Navy standard oil hose.

Cargo oil is discharged through two 11-inch connections in the pump room, leading to the 11-inch cross-connection on the shelter deck and the discharge mains on the upper deck, port and starboard. These mains are 11 inches diameter, extending aft along the upper deck from the pump room hatch for about 112 feet, where they rise to the shelter deck and are carried aft full size for a short distance and then reduced to 6 inches, continuing to the after end of the deck house abreast of the engine room. Deck discharge connections are taken off these mains. Forward of the pump room hatch the mains are

8 inches diameter for a distance of about 60 feet, where they branch into two 6-inch connections on each side of the vessel, one rising abruptly to the shelter deck and the other extending forward about 75 feet before rising to the shelter deck. All deck connections are flanged to suit 6-inch U. S. Navy standard oil hose.

The cargo tanks are flooded through two 10-inch sea valves in the pump room, either direct or through the pumps, and are pumped overboard via the shelter deck cross-connection. Four portable sections of 8-inch pipe of sufficient length to extend from the discharge outlets to the ship's side are provided for pumping overboard.

The forward fuel oil or ballast tanks are served by a fuel oil transfer pump located in the forward pump room. These tanks are provided with 6-inch deck filling connections, 5-inch suctions and a 4-inch discharge to the fuel oil transfer line and deck, the latter provided with a portable section of piping extending to the ship's side for overboard discharge.

All oil tanks are provided with vents, steam fire-extinguishing connections and steam heating coils.

DESCRIPTION OF MACHINERY.

The main propelling machinery is located in a common compartment in the after hold.

Main Engine.—The main engine is of the vertical, inverted, direct-acting, quadruple-expansion type. It has a stroke of 51 inches, and the following cylinder diameters: H.P., 24 inches; 1st I.P., 35 inches; 2nd I.P., 51 inches; and L.P., 75 inches. The engine turns to starboard when running ahead, and is designed to develop about 2,800 I.H.P., with steam at 220 pounds' pressure at the H.P. valve chest, when driving the vessel at the designed speed of 10.5 knots.

The engine is of conservative design, with cylinders of the usual construction; all fitted with working liners except the L.P. The cylinders are supported on cast iron columns of the

box type at the front and the inverted Y type at the back. Each column carries a crosshead guide. The bed plate is in four sections. It is of cast iron of box section, flanged and securely bolted together and to the engine foundation.

All reciprocating parts are of forged steel. Adjustable cast iron shoes, for working on the guides, are secured to each crosshead. The H.P. and 1st I.P. pistons are of cast iron; those for the 2nd I.P. and L.P. cylinders are cast steel. All pistons are fitted with cast iron junk rings. U. S. metallic packing is used for all piston rods and valve stems.

The oil system is through wick feed from brass manifold oil boxes.

The valve gear is of the Stephenson link type. All valves are fitted with balance pistons, except the H.P. piston valve. The valves and their settings are given in Table 1.

TABLE I.—VALVE SETTINGS.

	H. P.		1st I. P.		2d I. P.		L. P.	
Number	One		One		One		One	
Type	Piston		Piston		Slide		Slide	
Diameter, inches	11 and 11½		18½ and 19		.....		.....	
Travel, inches	8		8		8		9	
Takes steam	Inside		Outside		Outside		Outside	
	Top	Bot.	Top	Bot.	Top	Bot.	Top	Bot.
Width of port, inches	2½	2½	2½	2½	4¼×43	4¼×43	6×67	6×67
Steam opening, linear, inches	1 1/16	2 1/16	1 3/32	2 3/32	2 7/64	2 7/32	2 1/32	2 1/32
Steam opening, area, sq. ins.	54.25	54.65	87.04	92.75	181.4	190.8	330.81	339.18
Exhaust steam, linear, inches	2½	2½	2½	2½	4¾	4¾	6	6
Exhaust steam, area, sq. ins.	70	66.25	114	114	204.25	204.25	402	402
Steam lap, inches	2 1/16	1 5/16	2 3/32	1 3/32	1 3/32	1 3/32	2 1/32	1 1/32
Exhaust lap, inches	−1 1/32	− 1/32	− 1/32	0	− 1/32	0	− 1/32	+ 1/32
Steam lead, linear, inches	1/8	1/8	1/32	1/32	1/8	1/8	1/8	1/16
Cut-off, decimal of stroke	.73	.652	.728	.652	.742	.658	.743	.657

Connecting rod, length between centers, 115 inches. Ratio to crank 4.51.

Reversing Gear.—The reversing engine is secured to the bottom of the port column of the 2nd I.P. cylinder. It is of the ram type, with cylinder 14 inches diameter by 24 inches stroke. It is controlled from the working platform through the reversing lever and a floating lever attached to the reverse shaft. The reverse arms are slotted and fitted with screw blocks for independent linking up.

Turning Gear.—For jacking the main engine a vertical, single, reciprocating engine is provided. It is located on the after end of the main engine, port side. The steam cylinder is 8 inches diameter by 6 inches stroke.

Shafting and Bearings.—All shafting is forged steel of solid section, with flanged couplings forged on and secured together by forged steel bolts.

The crank shaft is in four sections of the built-up type. The four cranks are at equal angles, the sequence being H.P., 1st I.P., L.P., and 2nd I.P.

The propeller shaft is covered, within the stern tube, with composition sleeves ⅞-inch thick at the bearings and ½-inch thick between bearings. The sleeves are secured in the usual manner.

The shafting is of the following principal dimensions:

Crank shaft, length, feet and inches	29–8
diameter, inches	14¼
pins, length, inches	13
diameter, inches	14¼
Thrust shaft, length, feet and inches	11–6
diameter, inches	14¼
collars, number	10
thickness, inches	2¼
diameter, inches	23½
space between, inches	5
bearing surface, square inches	2742.6
Line shaft, length, feet and inches	13–0
diameter, inches	13½
Propeller shaft, length, feet and inches	17–3¼
diameter, inches	15

The thrust bearing is of the usual adjustable horse-shoe marine type.

The shaft bearings are of the following characteristics:

Crank shaft bearings:		
number		8
length, inches	(1) 13½, (6) 15 and (1)	16½
diameter, inches		14¼
bearing surface		White metal
Thrust shaft bearings:		
number		2
length, inches		14¼
diameter, inches		14¼
bearing surface		White metal
Thrust shoes, number		10
effective surface, square inches		1920
Line shaft bearings:		
number		2
length, inches		20
diameter, inches		13½
bearing surface		White metal
Stern tube bearings:		
Forward, length, inches		24
diameter, inches		16 13/16
bearing surface		Lignum vitae
After, length, inches		60
diameter, inches		16¾
bearing surface		Lignum vitae

Propeller.—The propeller is a right-hand true screw. It has four adjustable, detachable blades of manganese bronze. The hub and cap are cast iron. Each blade is secured to the hub by seven 3-inch steel studs and composition nuts. The hub has a taper fit on the propeller shaft, is securely keyed, and held on by a steel nut locked with a key.

PROPELLER DATA.

Diameter, feet and inches	17–6
Pitch as set, feet and inches	18–0

Pitch adjustable from 16 feet 8 inches to 19 feet 4 inches.

Pitch ratio	1.022
Projected area, square feet	85.8
Helicoidal area, square feet	101.4
Disc area, square feet	240.5
P.A. ÷ D.A.	0.3565
Height of lower tip of blade above keel, inches	13⅜
Immersion of upper tip of blade at load draught, inches	92⅞

Main Condenser.—There is one independent, cylindrical, double-flow, surface condenser located on the port side of the main engine. The shell and heads are of cast iron, tube sheets of rolled Muntz metal 1¼ inches thick, and seamless-drawn Admiralty metal tubes No. 16 B.W.G. thick, ¾-inch outside diameter and 8 feet 10⅝ inches long. The tubes are packed with corset lacing and brass ferrules. There are 2,471 tubes, giving a total cooling surface of 4,200 square feet.

Main Circulating Pump.—The circulating water is supplied by a double-suction, 14-inch, centrifugal pump, driven by a direct connected reciprocating engine having a cylinder 8 inches in diameter by 10-inch stroke. Two engines are provided, one as standby.

Main Air Pump.—The air pump is attached to the main engine and located abreast the L.P. cylinder, port side. It is of the vertical, double-acting, Edwards type, and has a 26-inch diameter cylinder with a 24-inch stroke.

Attached Bilge Pump.—There is one vertical bilge pump of the plunger type attached to the main air pump. It is 3½ inches diameter by 24-inch stroke.

Attached Sanitary Pump.—A vertical, sanitary pump, of same type and size as the bilge pump, is also attached to the main air pump.

Auxiliary Condenser.—A small auxiliary condenser of the Wheeler surface type, combined with horizontal air and circulating pump, is located in the engine room outboard of the main condenser. It has 577 tubes, ¾-inch outside diameter, No. 16 B.W.G. thick and 6 feet 5⅛ inches long, giving a cooling surface of 705 square feet.

TABLE II.—INDEPENDENT RECIPROCATING PUMPS.

No.	Service.	Size, Type and Make.	Location.
2	Main feed	12 in. × 8 in. × 24 in. vertical, single, double-acting, "Worthington."	Engine room
1	Auxiliary feed	12 in. × 8 in. × 24 in. vertical, single, double-acting, "Worthington."	Engine room
1	Donkey feed	8 in. × 5 in. × 12 in., horizontal, simplex, double-acting, "Worthington."	Engine room
1	Sanitary	6 in. × 5½ in. × 6 in., horizontal, duplex, double-acting, "Worthington."	Engine room
1	Auxiliary air and circulating	12 in. × 14 in. × 14 in. × 12 in., horizontal, simplex, double-acting, combined, "Worthington."	Engine room
1	Fresh water	4½ in. × 2¾ in. × 4 in., horizontal, duplex, double-acting.	Engine room
1	Evaporator feed	4½ in. × 2¾ in. × 4 in., horizontal, duplex, double-acting.	Engine room
2	Ballast, fire and bilge	8 in. × 8½ in. × 12 in., horizontal, duplex, double-acting, "Worthington."	1 in engine room 1 in forward pump room
2	Fuel oil service	5½ in. × 3½ in. × 5 in., horizontal, duplex, double-acting.	Boiler room
2	Fuel oil transfer	10 in. × 6 in. × 10 in., horizontal, duplex, double-acting, "National Transit Pump & Machine Co."	1 in boiler room 1 in forward pump room
2	Cargo oil	18 in. × 14 in. × 24 in., horizontal, duplex, double-acting, "National Transit Pump & Machine Co."	Pump room

Feed Water Heater.—In the engine room, outboard and above the feed pumps, is a No. 12 Reilly, type C, multicoil feed-water heater. Between the feed pumps and the feed heater is a grease extractor. Both the feed heater and the grease extractor can be by-passed.

Hotwell Tank.—One hotwell tank of about 500 gallons total capacity is installed. It is fitted with a float and chronometer valve gear for automatically regulating the feed pumps.

Independent Reciprocating Pumps.—Table II gives the principal data for all independent reciprocating pumps.

Boilers.—There are three main, horizontal, single-ended, Scotch boilers and one horizontal, single-ended, Scotch donkey boiler, all fitted with separate combustion chambers, and arranged to burn fuel oil. The main boilers are placed abreast just forward of the main engine, with fire room forward. The donkey boiler is located in the boiler hatch at the upper deck level.

BOILER DATA.

	Main (each).	Donkey.
Working pressure, pounds	220	180
Test pressure, pounds	330	270
Diameter, external, feet and inches	14–7 3/16	11–1
Length, over heads, feet and inches	11–6	10–10½
Furnaces, number	3	2
type	Morrison,	Suspension
diameter, internal, inches	43	39
length, external, feet and inches	8–1 13/16	7–9⅜
Heating surface, square feet	2345	1223
Tubes, material	Charcoal iron, lap welded	
number, ordinary	250	132
stay	100	56
outside diameter, inches	2¾	2¾
thickness, ordinary, B.W.G.	No. 10	No. 10.
stay, inches	5/16	5/16
length, ordinary, inches	93½	89
stay, inches	94¼	89¾
Safety valve, type	Twin, Ashton	
size	3	2½

Uptakes.—Uptakes of usual design for Howden, heated air, forced draft, connect the main boiler tube nests to the smoke pipe. For heating the air, 789 3-inch by No. 12 B.W.G. tubes, presenting a total heating surface of 2,169 square feet, are installed vertically in the uptakes. The gases of combustion pass through the tubes and the forced draft air outside.

The donkey boiler uptake connects into the smoke pipe immediately below the top of the boiler hatch.

Dampers are fitted in all uptakes.

Smoke Pipe.—There is one double casing smoke pipe, 8 feet 1 inch outside diameter and 80 feet high above the center of the lower furnaces. The clear cross-section area through the smoke pipe is 41.29 square feet.

The donkey boiler smoke pipe is inside the main smoke pipe and extends to the top. It is of single casing construction 33 inches inside diameter.

Forced Draft.—Howden, closed ash pit, heated air, forced draft is used for the main boilers. The air is supplied by one vertical, engine-driven, Sturtevant Type VS7, double-inlet, blower, with 110-inch diameter fan. The fan is provided with duplicate single engines, one being a standby. Each engine has a cylinder 7 inches diameter by 6 inches stroke, and is capable of driving the fan at 390 revolutions per minute. The unit is located in the engine room, starboard side, forward. The air is drawn from the engine room and delivered, through sheet-iron ducts of rectangular section, to the heater boxes in the uptakes, thence to the ash pits.

Fire Room Hoist.—A hoist of the steam ram type is fitted in the starboard ventilator to the fire room.

Fuel Oil System.—The Dahl system of mechanical oil-fuel burning is installed for both the main and donkey boilers. The apparatus comprises one oil burner per furnace, three oil heaters, two oil-fuel service and two oil-fuel transfer pumps, together with the necessary strainers, piping, etc. The service pumps are arranged to draw from all bunker tanks,

including summer tanks No. 5 and cofferdam No. 3, which are fitted for carrying bunker oil-fuel, and they discharge to the burners through the oil-fuel heaters or bypass same. All bunker tanks are fitted with high and low suctions.

Steam Piping.—The main steam pipe is of seamless drawn steel. The feeders from the boilers are 6 inches each, uniting into an 8-inch pipe leading to the main engine throttle valve. A 4½-inch auxiliary steam connection is taken off each main boiler, forming a main with branches to the engine room and deck, from which sub-branches are taken as required. The donkey boiler has a 4-inch connection to the auxiliary steam line. A 2½-inch independent steam line is led from the donkey boiler and No. 1 main boiler to the generator sets in the engine room; they also take steam from the auxiliary steam line in the engine room.

Auxiliary Exhaust Piping.—The exhaust steam from all auxiliaries is collected in a common main connecting to the main and auxiliary condensers, and the feed water heater. A back pressure valve is fitted at each condenser connection to maintain the necessary pressure on the line for the feed water heater. There is also a connection from the auxiliary exhaust line to the 2nd I.P. receiver.

Feed System.—Four feed pumps are provided as listed in Table II. All feed pumps take their suction from the hotwell tank. The main and auxiliary feed pumps are arranged to feed the main boiler direct or through the feed water heater. The donkey feed pumps feeds the donkey boiler direct or the main boilers via the auxiliary feed line. The feed stop and check valves are at the rear end of the boilers and are controlled from the engine room. One No. 13½ and one No. 10½ Metropolitan injector are also connected up to the feed lines.

Evaporating and Distilling Plant.—The plant consists of two Reilly multicoil evaporators, each having a capacity of 35 tons of fresh water per 24 hours, and two Reilly multicoil, vertical

distillers, each of 6,000 gallons' capacity per 24 hours, together with the necessary pumps as listed in Table II. The evaporators are located in the engine room, at the working level, starboard side, aft, with the distillers above at high elevation in the engine hatch.

Refrigerating Plant.—The ice machine room is starboard of the engine room, and it is entered from the middle grating. The outfit includes a 2-ton Kroeschell, vertical, double acting, CO_2 compressor driven by a direct-connected steam engine, oil separator, CO_2 receiver and condenser, together with the usual appurtenances.

The refrigerating rooms are abaft the engine room on the main deck.

Lighting Plant.—There are two G. E. Co. direct current, 20-KW, horizontal, compound-wound, Type MP6-20-400, generators, each driven by a 9-inch × 7-inch vertical, single, direct-connected, steam engine. They are located on a flat in the after end of the engine room. The switchboard is just abaft the generators. Current is supplied at 110 volts.

An emergency generator-set is also provided. It is located on the upper deck forward and consists of a Westinghouse, direct current, 12.5 kilo-watt, 125 volts, generator, designed for 600 revolutions per minute, and driven by a 25 horsepower, four-cylinder, Clifton Motor Works kerosene engine.

Workshop.—A small workshop is provided starboard of the engine room, at the level of the middle grating. It is equipped with the following tools and equipment:

1 16-inch Steptoe shaper.
1 Buffalo Forge Co. hand drill.
1 12-inch Blount, double-energy grinder.
1 Grindstone.
1 16-inch Davis lathe.
1 20-inch Fairbanks upright drill.
1 Anvil.
1 Portable forge.

1 Work bench.
2 Vises.
Outfit of hand tools.

All tools are belt driven from overhead shaft and pulleys. The power is supplied by a 7 H.P., G. E. Co., direct-current motor, designed for 1,150 revolutions per minute.

TRIALS.

The contract required:

(*a*) A trial over a measured mile in the deepest water available and at the highest speed obtainable, not less than three runs over the mile to be made, and the mean speed and revolutions for these runs carefully ascertained.

(*b*) A full speed trial of four hours' duration in deep water at the highest speed attainable (not less than 10½ knots), to be determined by the average revolutions of the main shaft according to the measured mile trial. During this trial all the auxiliaries necessary for the service of the vessel, including dynamos for efficient lighting, shall be in operation. The fuel oil consumption on this trial to be measured and not to exceed the equivalent of 325 pounds per knot at 10½ knots.

(*c*) Turning and backing trials to test the engines and steering gear.

The *Ramapo* sailed from Newport News for her trials on the morning of October 15, 1919, and proceeded to the Delaware Breakwater, where the standardization runs were made over the measured mile.

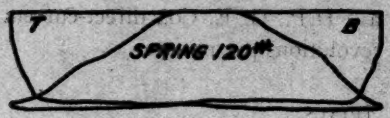
The four-hour official full-power trial was conducted on the run up the coast to the Delaware Breakwater. At the builders' request, the oil and water consumption, the latter not required by the contract, were measured as follows: During the first three hours of the official trial the total oil consumption and the water consumption of the main engine and auxiliaries was measured. Then followed two hours at full power measuring

U.S.S. RAMAPO

Full Power Trial Indicator Cards of Set Taken at 2:45 P.M., Oct. 15, 1919.

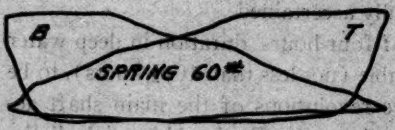
R.P.M. 74.9 – I.H.P. 2839

H.P. Cyl.



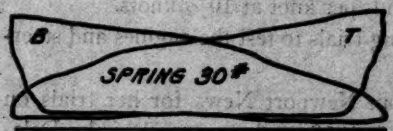
M.E.P.	84#
M.R.P.	8.36#
I.H.P.	710
St. Chest	222#
Cutoff	67%

1st I.P. Cyl.



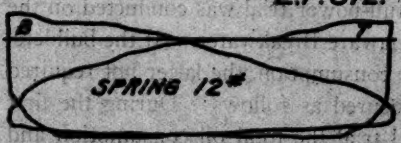
M.E.P.	39.3#
M.R.P.	8.46#
I.H.P.	719
Rec'r.	99#
Cutoff	68.7%

2nd I.P. Cyl.



M.E.P.	18#
M.R.P.	8.3#
I.H.P.	705
Rec'r.	37#
Cutoff	68.8%

L.P. Cyl.



M.E.P.	8.3#
I.H.P.	705
Rec'r.	7#
Vacuum	27.5"
Cutoff	68%

PLATE I.

Table III - U.S.S. RANAPO - Trial Data.

	Full Power Trials.			Reduced Power Trials.	
	4 Hours Official	8 Hours Measuring Oil,& Water M.E.& Auxs.	8 Hours Measuring Oil,& Water M.E.only.	4 Hours Measuring Oil,& Water M.E.& Auxs.	2 Hours Measuring Oil,& Water M.E.only.
Date of trial	Oct.15,1919	Oct.15,1919	Oct.15,1919	Oct.16,1919	Oct.16,1919
Time	11:45 A.M. to 3:45 P.M.	11:45 A.M. to 2:45 P.M.	8:15 P.M. to 4:15 P.M.	2:15 P.M. to 6:15 P.M.	6:30 P.M. to 8:30 P.M.
Speed in knots (from curve)	11.2	11.19	11.22	9.09	9.14
Slip of propeller, per cent.	15.14	15.1	15.22	12.27	12.53
Displacement, corresponding, tons	16648	16648*	16648*	16622*	16622*
Pressures (average):					
Main steam, boilers, lbs. gauge	222	222	222	220	220
H.P. chest, " "	218	218	217	216	216
1st rec'r. " "	100	101	95.8	78	77
2nd " " "	35.4	35.4	26.4	25.6	24.1
L.P. " " "	7	7	6.75	2.24	2.28
Vacuum, main cond., inches of mercury	27.6	27.5	27.9	28	28
Barometer, inches of mercury	30.48	30.48	30.48	30.28	30.28
Main feed, lbs. gauge	245	244	245	247	248
Auxiliary exhaust, lbs. gauge	12	12	11.6	10	9
Fuel oil, at burners, lbs. (average)	160	164	182	98	106
Forced draft, inches of water	1.27	1.24	1.38	0.5	0.4
M.E.P., lbs. per sq.in. (average):					
H.P. cylinder	35.7	34	35.4	21.4	23.1
1st I.P. cylinder	21.4	21.5	20.7	23.9	25.8
2nd I.P. "	18.7	18.5	19.3	14.6	14.9
L.P. "	9.09	8.98	9.42	6.09	6.17
Mean referred pressure	24.8	24.7	24.7	24.4	24.9
R.P.M. or double strokes per min. (average):					
Main engine	74.8	74.2	74.5	59	59.5
" circulating pump	111	109.7	116.5	112	110
" feed pump	12	12.2	11.8	5	7
Forced draft blowers	252	250	256	174	168

2nd I.P. "	18.7	18.5	19.5	20.5	20.7
L.P. "	9.00	8.98	9.42	6.00	6.17
Mean referred pressure	24.8	24.7	24.7	24.4	24.9
R.P.M. or double strokes per min. (average):					
Main engine	74.5	74.2	74.5	58	59.5
" circulating pump	111	109.7	118.5	112	110
" feed pump	12	12.2	11.8	5	7
Forced draft blower	282	280	276	174	248
Fuel oil pumps	11.7	11.8	14	8	9
I.H.P., main engine (average):					
H.P. cylinder	705	705	704	405	419
1st I.P. "	725	726	745	372	393
2nd I.P. "	728	718	754	447	449
L.P. "	784	785	799	406	412
Total	2942	2930	2990	1630	1685
Temperatures, degrees F. (average):					
Main injection	69.6	70.5	67.5	68	68
" overboard discharge	99.8	101.3	94	80	79
Air pump discharge	110.5	114.6	97.7	87	85
Hotwell	118.5	125.2	101.5	97	84
Main feed	190	196	176	188	192
Air to forced draft blower	102.5	104	102.5	90	90
Oil at burners	145.8	144.5	139.5	139	145
Smoke pipe	548	545	565	424	450
Engine room working level	98.5	98.6	98.5	88	88
Fire room " "	110	110	109	86	86
Water consumption:***					
Pounds per hour, main engines	---	---	58251	---	28198
" " and auxs.	---	48161	---	28135	---
per I.H.P. main engines	---	14.4	12.8	14.16	12
evaporated, actual	---	45701	---	24673	---
per sq.ft.of H.S.	---	6.25	---	12	---
per lb.of oil, actual	---	15.5	---	15.5	---
equiv.	---	14.4	---	14.58	---
Fuel oil consumption:***					
(Oil analysis: 19039 B.T.U. per pound, Specific gravity at 60° F. 0.92, Baume degrees at 60° F. 25, flash point 220° F.)					
Pounds per hour	---	8246	2550	1887	1841
knot run	---	289	297	198	198
at 10.5 knots	---	275	---	---	---
guaranteed	---	325	---	---	---
below guarantee	---	52	---	---	---
hour per I.H.P. main engines	---	1.1	1.113	1.107	1.079
sq. ft. of H.S.	---	0.462	0.477	**	0.393
Miscellaneous data:					
Per cent. CO_2	9.8	10	9.55	10.5	9.7
Number of boilers in use	3	3	3	3 for 1½ hrs. 2 " 3½ "	2
" " burners in use	9	9	9	4 to 6	4 to 6
Main engine linked up, inches	---	1-0-0-1¼	1-0-½-1½	8-2-2-8	8-2-2-8

* Approximate.
** One boiler cut out during run.
*** All water and oil consumptions corrected for extra pumps used for measuring water and oil.

the total oil consumption and the water consumption of the main engine only, other conditions remaining unchanged. This was done during the last half hour of the official trial and continued an hour and a half thereafter. The data obtained is given in Table III.

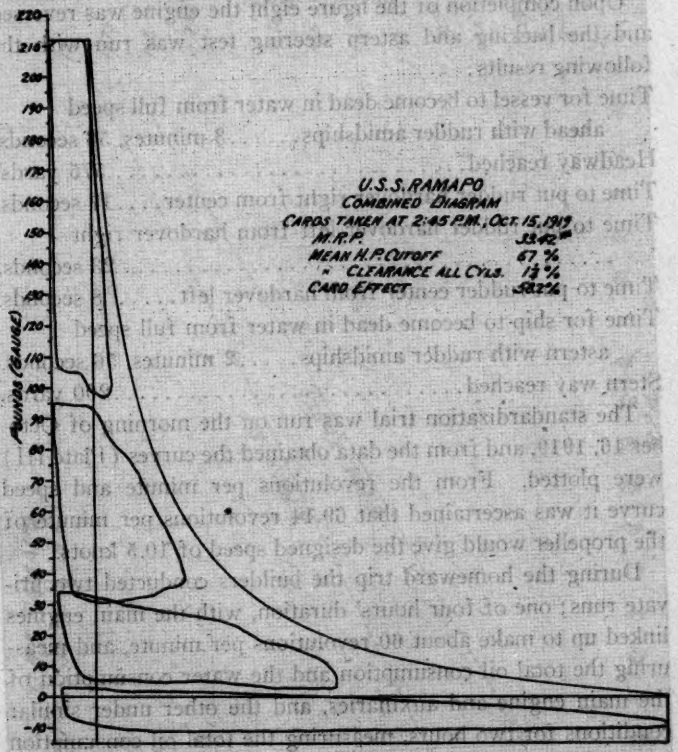


PLATE II.

Plate I shows a set of indicator cards taken during the official trial and Plate II shows the combined card of same set.

Following these trials the steering and backing tests were conducted. While making 70 revolutions per minute ahead

the vessel was swung through a figure eight course. Time required to put rudder hardover right from center was 12 seconds, estimated diameter of turning circle 650 yards. Time required to put rudder hardover left from hardover right was 22 seconds, estimated diameter of turning circle, 550 yards.

Upon completion of the figure eight the engine was reversed and the backing and astern steering test was run with the following results:

Time for vessel to become dead in water from full speed ahead with rudder amidships.....3 minutes, 56 seconds.
Headway reached.....575 yards.
Time to put rudder hardover right from center....17 seconds.
Time to put rudder hardover left from hardover right .....23 seconds.
Time to put rudder center from hardover left..... 8 seconds.
Time for ship to become dead in water from full speed astern with rudder amidships.....2 minutes, 56 seconds.
Stern way reached.....200 yards.

The standardization trial was run on the morning of October 16, 1919, and from the data obtained the curves (Plate III) were plotted. From the revolutions per minute and speed curve it was ascertained that 69.14 revolutions per minute of the propeller would give the designed speed of 10.5 knots.

During the homeward trip the builders conducted two private runs; one of four hours' duration, with the main engines linked up to make about 60 revolutions per minute, and measuring the total oil consumption and the water consumption of the main engine and auxiliaries, and the other under similar conditions for two hours, measuring the total oil consumption and the water consumption of the main engine only. The data from these trials is also given in Table III.

The data given in Table III for the official four-hour trial is that taken during the trial by the builders and should not differ from the official figures by any appreciable amount.

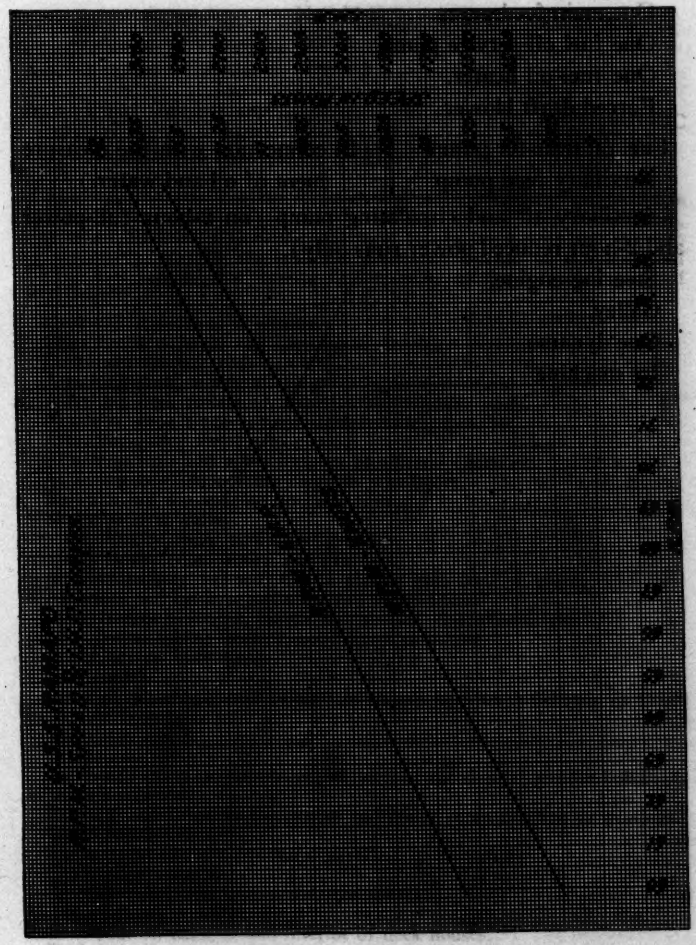


PLATE III.

The following auxiliaries were in use on all the trials:

Main circulating pump.
One main feed pump.
One fuel oil service pump.
One fuel oil heater.
Forced draft blower.
Fuel oil transfer pump. } (In connection with the measure-
Auxiliary feed pump. } ment of oil and water.)
Auxiliary air and circulating pump (on 2-hour full-power and 2-hour reduced power runs only).
Steering engine.
Galley.
One dynamo.
Ice machine.

NOTES.

PARTS OF VESSEL TO WHICH APPLICATION OF ELECTRIC WELDING HAS BEEN CONSIDERED AND APPROVED BY AMERICAN BUREAU OF SHIPPING AND LLOYDS.

Although the classification societies are proceeding very cautiously in approving the application of electric welding to new ship construction, because of the lack of a reliable system of testing the quality and strength of the welds, nevertheless both the American Bureau of Shipping and Lloyds now approve the welding by electricity of the following parts of steel vessels:

Deck rail stanchions to plating.
Clips for detachable rail stanchions.
Continuous railing rods (joints).
Attaching deck collars (L rings) around ventilators.
Attaching deck collars (L rings) around smoke stack.
Attaching cape rings around smoke stack, pipes, etc.
Attaching galley fixtures to plating.
Attaching bath and other fixtures in officers' quarters.
Attaching cowl supporting rings to ventilators.
Bulwark rail top splicing and end fitting.
Skylights over galley.
(*a*) Engine room stairs and gratings.
(*b*) Boiler room stairs and gratings.
Attaching (*a*) and (*b*) to plating grab rods on casing.
All stairs and ladders, including rail attachments.
Door frames to casing, hinges, catch holds, coachhooks, etc.
Clips for attaching interior wood finish to casing.
Entire screen bulkhead.
Coal chutes.
Butts of watertight and oiltight boundary bars on bulkheads or floors in double bottom.
Ventilator cowls.
Stacks and uptakes.
Bulkheads (that are not structural parts of the ship), partition bulkheads in accommodation.
Framing and supports for engine and boiler room flooring or gratings.
Cargo batten cleats.
Tanks (that are not structural parts).
Shaft alley escapes.
Steel skylights over accommodation spaces.
Engine room skylights.
Grab rods on exterior and interior of deck houses.
Deck houses not covering unprotected openings through weather decks.
Reinforcing and protecting angles around manholes.
Joints of watertight angle collars at frames in way of watertight flats.—"International Marine Engineering."

THE EVOLUTION OF THE DESTROYER.

REMARKABLE DEVELOPMENT OF PRESENT-DAY DESTROYER FROM EARLY TYPE TORPEDO BOAT—RAPID ADVANCES IN BOILER AND TURBINE CONSTRUCTION.

BY COMMANDER S. M. ROBINSON, U. S. N.

Progress in marine engineering has been very rapid during the past thirty years, but it is believed that the development of the torpedo boat has been more rapid than that of any other type of vessel. Undoubtedly the main reason for this rapid progress is that it takes a very short time, comparatively, to build a destroyer, so that the engineer can test his ideas and make several advances in the time required to build one battleship. It is certainly a far cry from the 22.5-knot *Cushing* to the 36.88-knot *Dent*. The *Cushing* was our first torpedo boat and the *Dent* is one of our latest destroyers. The following table and accompanying photographs give a comparison between the two boats:

	Cushing.	*Dent.*
Length overall.....	137 feet 6 inches	314 feet 4½ inches
Beam, extreme.....	15 feet ½ inch	30 feet 1½ inches
Draft .....?.....	4 feet 6 inches	9 feet 0¾ inch
Displacement	91.34 tons	1,159 tons
Fuel	33 tons coal	288 tons oil
Speed	22.5 knots	36.88 knots
Horsepower	1,720 (indicated)	28,190 (shaft)
Revolutions per minute.	370	467.5
Number of screws.....	2	2
Cruising radius at 10 knots' speed.....	990 knots	at 15 knots, 4,960 knots
Boilers	2 Thornycroft	4 White-Forster
Total heating surface..	4,750 square feet	27,000 square feet
Total grate surface.....	76.6	
Ratio of heating surface to grate surface.....	62 to 1	

The *Cushing* was our first torpedo boat and was designed and built in 1890 by Nathaniel Herreshoff at the Herreshoff Works, Bristol, R. I. Inasmuch as she represents our maiden effort in this line of marine engineering, a brief description of her machinery may be of interest.

MACHINERY OF THE "CUSHING."

She was equipped with twin screws, each driven by a five-cylinder, quadruple expansion engine. The cylinders were 11½, 16, 22½, 22½ and 22½ inches in diameter, respectively. The stroke was 15 inches. The engine supports were a distinct advance in engineering for that day and consisted of 1½-inch steel rods, braced diagonally, and forming the cap bolts of the main bearings; the latter were secured to a bed-plate consisting of a single sheet of ¾-inch wrought iron with openings cut for the cranks. The bed-plate was secured in a fore-and-aft direction to keelsons and was also supported under the main bearings of the high-pressure and after low-pressure cylinders, the other main bearings being entirely without support; but, owing to the extreme care taken in balancing, the engines ran without vibration.

The boat was equipped with two Thornycroft boilers each having 38.3 square feet of grate surface and 2,375 square feet of heating surface. The boilers also represented an advance in engineering, as the longitudinal seams in the drums were welded instead of riveted. The boilers were designed for a pressure of 250 pounds per square inch.

For some time after the trials of the *Cushing*, the advance in torpedo boat engineering was limited by the progress in the design of propellers and the art of balancing engines. Like many other points of engineering, the screw propeller was not as well understood at that time as it is today, and many failures on trials were recorded due solely to improper propeller design. Also the higher speed of ship desired required higher engine speeds than had been the custom, with the result that vibration was so bad in many cases as to make it impossible to run at full power. However, both of these problems were successfully solved and the art of engine balancing, in particular, was carried to a high degree of success. This line of development reached its climax with the *Stewart* class, which were built in 1902.

THE LAST OF THE RECIPROCATING ENGINE BOATS.

The following table gives the general characteristics of these boats:

Length	245 feet
Beam	23 feet ¼ inch
Draft	6 feet 1¼ inches
Displacement	444 tons
Speed	29.3 knots
Indicated horsepower	8,000
Revolutions per minute	330
Number of screws	2
Fuel, coal	180 tons
Boilers	4 Thornycroft
Total heating surface	17,770 square feet
Total grate surface	237 square feet
Ratio of heating surface to grate surface	56.4 to 1
Radius of action at 12 knots	2,160 knots

These were the last of the reciprocating engine boats. We see that the speed has increased from 22.5 knots to 29.3 knots, the boiler pressure from 250 to 300 pounds, the indicated horsepower from 1,720 to 8,000, and the high-pressure cylinder from 11½ inches to 23 inches diameter. There is no doubt but what with the present-day methods of lubrication and other improvements of design, such as superheat, etc., destroyers of higher speeds could be built using reciprocating engines, but the improvement would not be great and the destroyer had just about reached its limitation as to speed.

Up to this time torpedo boats and destroyers had been regarded as coast and harbor defense vessels and long-distance cruising had not been carried out by them. The ordinary method of operation was to make short runs at high speed from a base, but in 1904 two of these boats, the *Preble* and *Paul Jones*, were sent to Panama to act as despatch vessels, and later in the year a flotilla of destroyers was sent from the Atlantic coast to the Asiatic station, via the Mediterranean. Both cruises proved conclusively that the destroyer was a reliable sea-going vessel and had a cruising radius that compared favorably with other types of ships. The fuel economy at low speeds was the greatest surprise of all. At that time so little was known regarding the fuel consumption at cruising speeds that

when the cruise from San Francisco to Panama was first projected arrangements were made for coaling at San Diego and every few hundred miles from there to Panama. Great was the surprise when it was found that the trip could easily be made with only one stop and without any stop by taking a small deck-load of coal.

ADOPTION OF THE TURBINE FOR PROPULSION.

The next step in destroyer design was the change to turbines for propulsion, and with this came the use of forced feed for lubrication. The first vessels to have this type of machinery were the *Smith* class, which were built in 1909. The following table gives the characteristics of these boats:

Length	293 feet 10½ inches
Beam	26 feet 4½ inches
Draft	8 feet ½ inch
Displacement	716 tons
Speed	28.35 knots
Shaft horsepower	9,946
Revolutions per minute	724
Number of screws	3
Boilers	4 Mosher
Total heating surface	18,003 square feet
Total grate surface	368.5 square feet
Ratio of heating surface to grate surface	48.86 to 1
Fuel, coal	304 tons
Cruising radius at 16 knots	2,800 knots

With previous destroyers fitted with reciprocating engines it had been difficult to obtain machinery that would be reliable for high-speed runs for any considerable period of time. The difficulty was so great that most of the early destroyers were required to run full-speed trials of one-hour duration only. As soon as turbines were adopted, the trial requirements were immediately raised to four hours. The destroyer, being a high-speed boat, requires high speed of revolutions of the screw, and this condition, while ideal for turbines, is a very severe test for reciprocating engines. There is no doubt but what the destroyer was the main factor in developing turbines for marine purposes, as it presented a case where turbines were fairly well suited for the purpose without any great change from conditions on shore. This was not the case with battleships and other vessels having slow-speed screws, and the progress of turbines for marine propulsion would doubtless have been very slow if it had not been for the destroyers.

BLOWER TROUBLES ELIMINATED BY TURBINE DRIVE.

Up to this time the forced draft blowers of destroyers had all been driven by light, high-speed, reciprocating engines and they were subject to the same troubles as were the main reciprocating engines, except that the troubles were exaggerated in the smaller engines. It was only natural, therefore, to adopt the turbine for driving the blowers as well as the propellers. It is no exaggeration to say that this change was almost as great an improvement as the adoption of the turbine for propulsion, as more breakdowns and failures on trials were due to blower trouble than to all other causes.

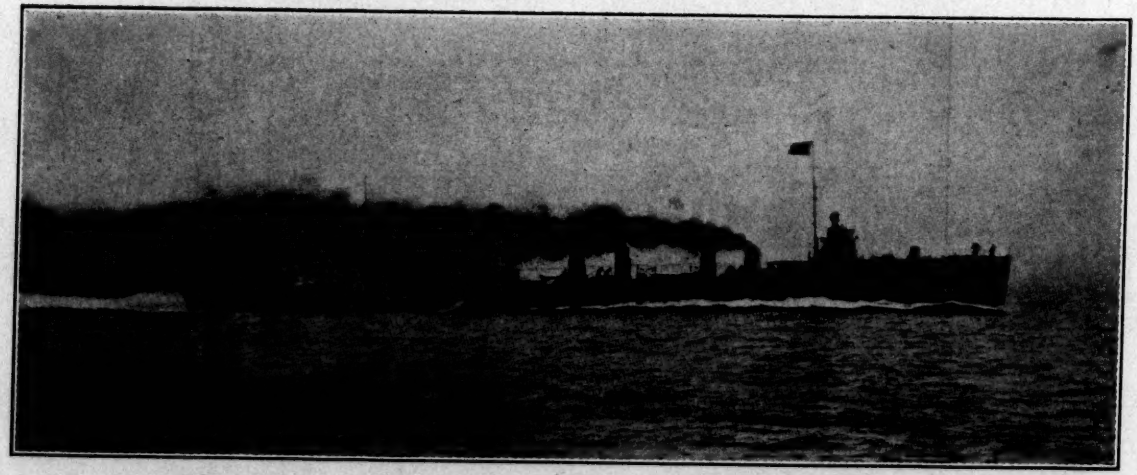


FIG. 1.—"SMITH" CLASS DESTROYER. THESE VESSELS, BUILT IN 1909, WERE THE FIRST AMERICAN TURBINE-DRIVEN DESTROYERS.

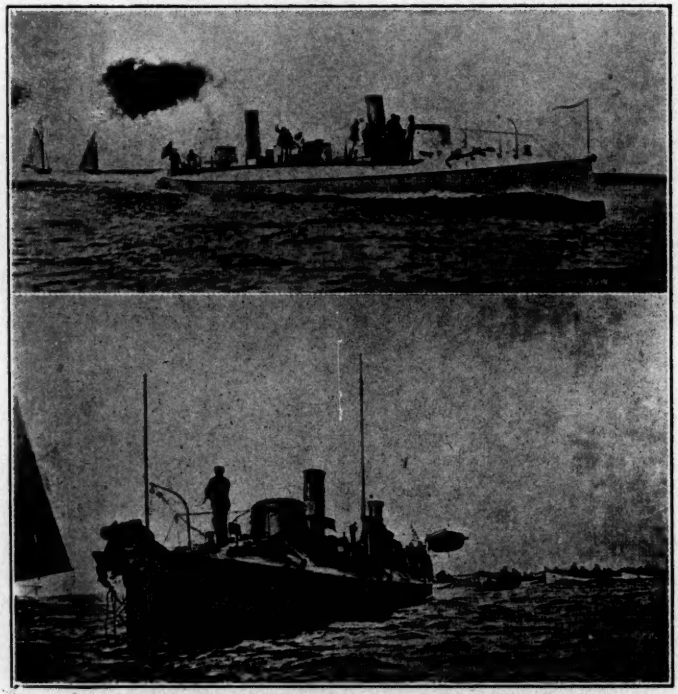


FIG. 2.—THE "CUSHING," THE FIRST UNITED STATES TORPEDO BOAT.
FIG. 3.—THE "CUSHING," WITH A DISPLACEMENT OF 91.34 TONS AND AN INDICATED HORSEPOWER OF 1,720, DEVELOPED A SPEED OF 22.5 KNOTS.

FIG. 4.—DESTROYER "ROE," BUILT AT NEWPORT NEWS IN 1910; TRIAL SPEED, 29.60 KNOTS.

FIG. 5.—"CHAUNCEY" CLASS DESTROYER (1900-1902).

FIG. 19.—U. S. S. "DENT," LATEST TYPE AMERICAN DESTROYER; DISPLACEMENT, 1,159 TONS; SHAFT HORSEPOWER, 28,190; SPEED, 36.88 KNOTS.

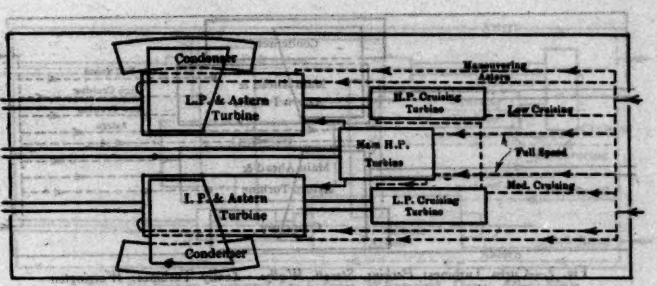


Fig. 6.—Parsons Turbines: *Smith* Class (Twenty Vessels)

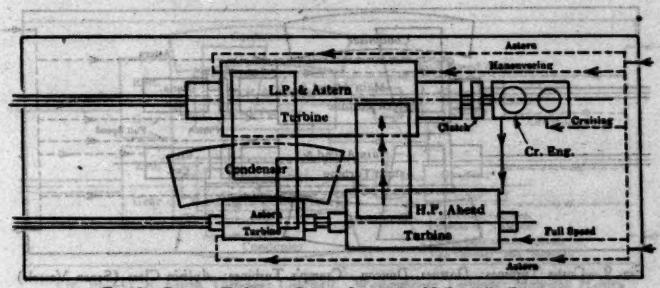


Fig. 8.—Parsons Turbines: *Cassin, Cummings, McDougall, Ericsson*

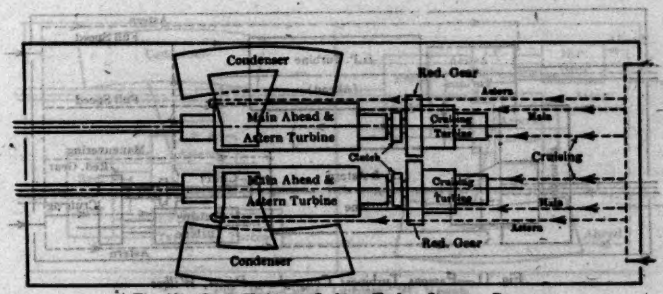


Fig. 10.—Curtis Turbines: *Cushing, Tucker, Sampson, Rowan*

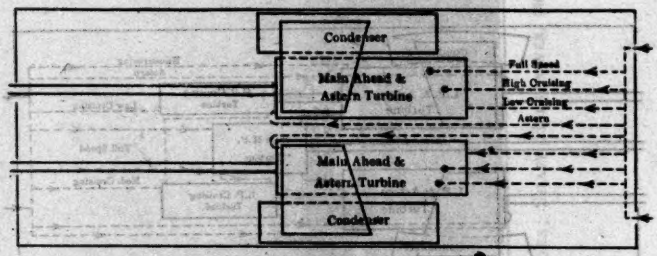


Fig. 7.—Curtis Turbines: *Parkin, Stevtt, Wolfe.* Zoelly Turbines: *Warrington*

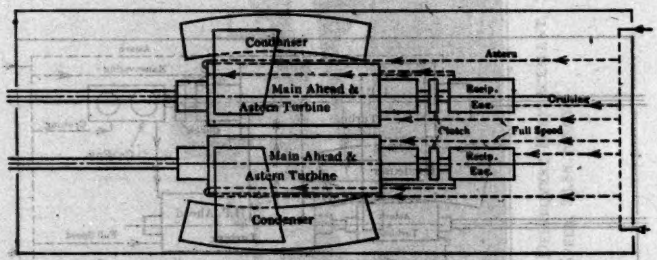


Fig. 9.—Curtis Turbines: *Downes, Duncan.* Cramp's Turbines: *Aylwin* Class (Seven Vessels)

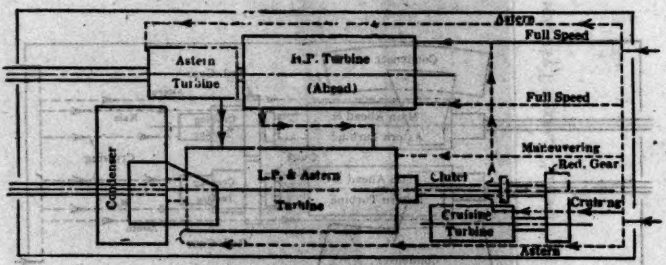


Fig. 11.—Parsons Turbines: *Campbeltown, Porter, Wilkes*

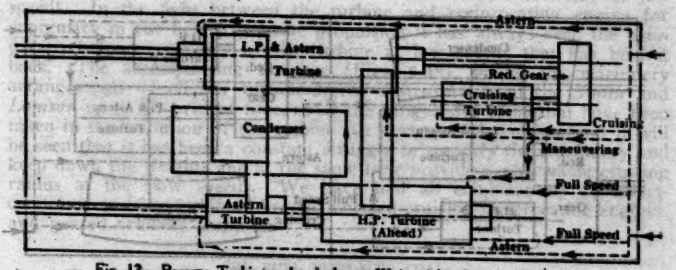


Fig. 12.—Parsons Turbines: *Jacob Jones, Wainwright, Davis, Allen, Shaw*

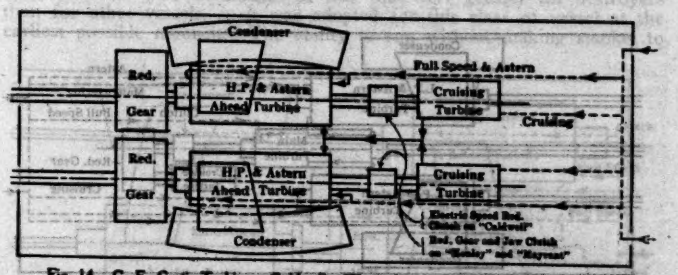


Fig. 14.—G. E. Curtis Turbines: *Caldwell*. Westinghouse Turbines: *Maynard, Henley*

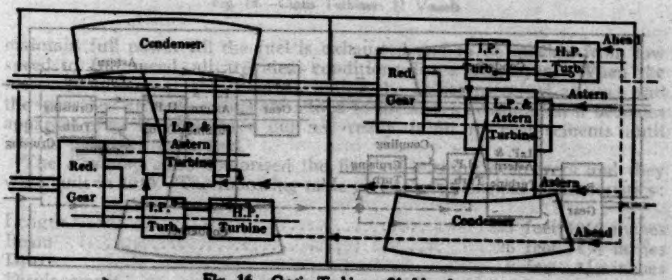


Fig. 16.—Curtis Turbines: 91 Vessels

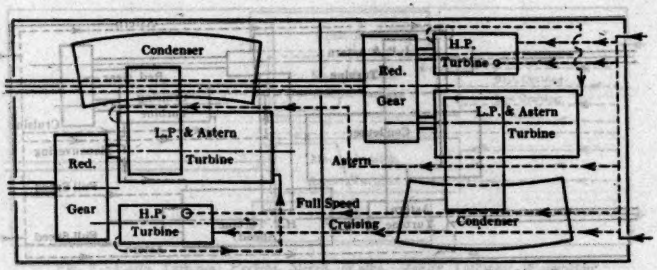


Fig. 13.—Parsons Turbines: *Wadsworth* Class (89 Vessels). Westinghouse Turbines: *Wadsworth* Class (40 Vessels)

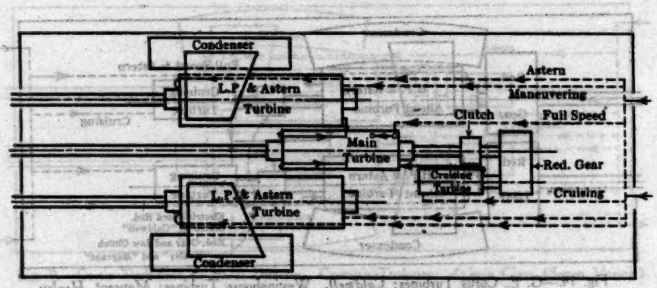


Fig. 15.—Parsons Turbines: *Conner, Stockton*

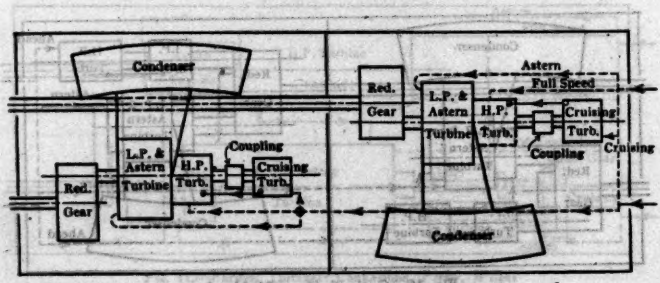


Fig. 17.—G. E. Curtis Turbines: 46 Vessels

The first turbine-driven destroyers solved the problem of reliable high-speed machinery, but there was very little saving in weight, owing to the complications that were necessary to obtain reasonable economy at low speeds. In the fight between the turbine and reciprocating engine for supremacy in the field of marine propulsion it has always been the comparatively poor economy of the turbine at low speeds that has held it back. The accompanying sketches (Figs. 6-18), giving the machinery arrangements of all our turbine-driven destroyers from the *Smith* and *Lamson* up to the present time, show the progressive steps that have been taken in the evolution of the turbine for destroyer propulsion, and it will be seen that it has been a constant struggle to simplify the machinery and keep down the weights and at the same time provide a reasonable cruising radius at the slow speeds. We have had all combinations of direct-connected cruising turbines, direct-connected cruising reciprocating engines, and geared cruising turbines.

OIL FUEL DISPLACES COAL.

The next advance in destroyer engineering was the substitution of oil fuel for coal. The advantages of this fuel were greater for destroyers than for other vessels, so it was adopted for this class of vessel at the earliest possible moment. The ability to run without making smoke, to

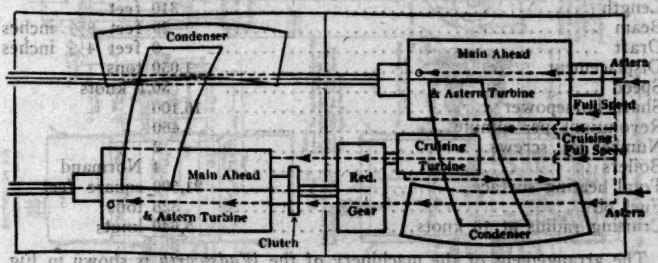


Fig. 18.—Curtis Turbines: 11 Vessels

maintain full power till the fuel is exhausted and to go quickly from low speed to full speed, all are ideal conditions for a destroyer. When the increased facility of refueling (including the ability to refuel at sea) and the increased cruising radius are added to the above advantages, it becomes apparent that the destroyer did not really fulfill its requirements until it was given oil as a fuel.

The *Paulding* class comprised the first oil-burning destroyers and they were built in 1910. The following table gives their principal characteristics:

Length	293 feet 10½ inches
Beam	26 feet 4½ inches
Draft	8 feet 1¼ inches
Displacement	711 tons
Speed	32.8 knots
Shaft horsepower	17,393
Revolutions per minute	903
Number of screws	3
Boilers	4 Normand

Total heating surface	19,321 square feet
Fuel oil	241 tons
Cruising radius at 16 knots	3,000 miles

While the propeller speeds of the destroyer were such as to allow better turbines than in the case of other ships, the speeds were not high enough for the best turbine practice, and in the attempts to improve this condition the propeller speeds used were considerably higher than should have been to get the best propeller efficiency. It was therefore natural to turn to the reduction gear as soon as it became sufficiently developed for the purpose. Here again the destroyer was the pioneer and led the way in the use of the mechanical reduction gear. This was natural, since the speed reduction was not nearly so great as with other types of ships (being only about 5 to 1), and therefore was a very much easier problem to solve; also the advantages gained were much greater than with other types of ships, owing to the great advantages of weight saving in this type of vessel.

REDUCTION GEARS APPLIED TO MAIN PROPELLING UNITS.

The *Wadsworth*, built in 1915, was the first vessel to be fitted with reduction gear for the main unit. The following table gives the general characteristics of the vessel:

Length	310 feet
Beam	29 feet 8½ inches
Draft	9 feet 4½ inches
Displacement	1,050 tons
Speed	30.7 knots
Shaft horsepower	16,100
Revolutions per minute	460
Number of screws	2
Boilers	4 Normand
Total heating surface	21,500 square feet
Fuel oil	326 tons
Cruising radius at 16 knots	5,640 knots

The arrangement of the machinery of the *Wadsworth* is shown in Fig. 13, and it will be noted that it is very simple, consisting of a high-pressure and a low-pressure turbine driving pinions meshing with the same gear, there being only two gears and two main shafts. The *Wadsworth's* machinery contains several novel features. The gear is a single helical gear and is arranged to take part of the propeller thrust direct and transmit it to the turbines, where it is balanced by steam pressure; the remainder of the thrust is taken by a thrust bearing which is of the pivoted segmental (Kingsbury) type. This was the first use of this type of thrust bearing in the navy, but its use has since become practically universal. The single helical gear has not been used in the later destroyers, which are of much greater horsepower (requiring longer pinions), but it has proved very satisfactory on the *Wadsworth*.

The *Wadsworth* really marked the end of the era of direct-connected turbines, although several destroyers were built after this date with direct-connected turbines. During the war it was not possible to obtain gears in such great numbers as they were needed, so a certain number of direct-connected turbines were installed. These reached a high degree of perfection in the boats turned out by Newport News Shipbuilding & Drydock Company, as these boats made over 35 knots on their full power trials.

BOILER CAPACITY INCREASED TO OBTAIN HIGHER SPEEDS.

The next step in the evolution of the destroyer was to increase the speed by a large increase in boiler capacity. These boats were built during the war under great pressure and various expedients were considered in the hope of expediting production. It was even considered reducing the

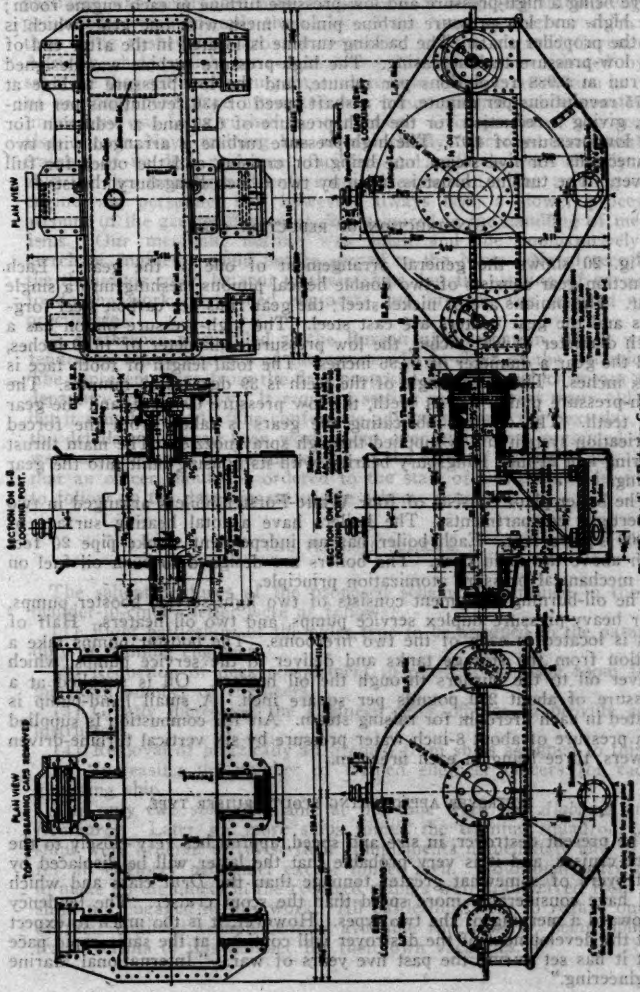


Fig. 28.—Detail of Reduction Gears

speed, reducing the number of boilers, etc., but none of these offered any hope of greatly increased production, so it was decided to build the best boat possible under the conditions. The data on this class of vessel are given in the first page of this article in comparison with the *Cushing*.

The propelling machinery is located in two separate compartments, one forward of the other. It consists of geared turbines of the Parsons type, there being a high-pressure and low-pressure turbine in each engine room; the high- and low-pressure turbine pinions mesh with one gear, which is on the propeller shaft. The backing turbine is located in the after end of the low-pressure turbine casing. The high-pressure turbine was designed to run at 2,988 revolutions per minute, and the low-pressure turbine at 1,775 revolutions per minute, for a shaft speed of 436 revolutions per minute, giving a reduction for the high pressure of 6.85 and a reduction for the low pressure of 4.07. The high-pressure turbine is arranged with two connections for live steam, one being for cruising and the other for full power. The turbine thrust is taken by two 9-inch Kingsbury thrusts.

CONSTRUCTION OF REDUCTION GEAR.

Fig. 20 shows the general arrangement of one of the gears. Each reduction gear consists of two double helical pinions meshing into a single gear. The pinions are of nickel steel; the gear rims are carbon steel forgings and the gear centers are cast steel. The high-pressure pinion has a pitch diameter of 9.91 inches, the low pressure a diameter of 16.69 inches, and the gear a diameter of 67.96 inches. The total length of tooth face is 37⅞ inches. The spiral angle of the teeth is 39 degrees 53 minutes. The high-pressure pinion has 41 teeth, the low pressure 69 teeth, and the gear 281 teeth. The oil for lubricating the gears is taken from the forced lubrication system and is supplied through spray nozzles. The main thrust bearing is a 21-inch Kingsbury bearing with its housing built into the gear casing.

The boiler plant consists of four White-Forster boilers arranged in two watertight compartments. The boilers have a total heating surface of 27,000 square feet. Each boiler has an independent smoke pipe 20 feet high above the main deck. The boilers are designed to burn oil fuel on the mechanical pressure atomization principle.

The oil-burning equipment consists of two light-service booster pumps, four heavy-pressure duplex service pumps, and two oil heaters. Half of this is located in each of the two firerooms. The booster pumps take a suction from the storage tanks and deliver to the service pumps, which deliver oil to the burners through the oil heaters. Oil is supplied at a pressure of about 250 pounds per square inch. A small hand-pump is located in each fireroom for raising steam. Air for combustion is supplied at a pressure of about 8-inch water pressure by six vertical turbine-driven blowers, three being in each fireroom.

DESTROYER APPROACHING SCOUT CRUISER TYPE.

The present destroyer, in size and speed, approaches very closely to the scout cruiser, and it is very probable that the latter will be displaced by destroyers of somewhat greater tonnage than the *Dent* class and which will have considerably more speed than the scout cruiser. The tendency is toward a merging of the two types. However, it is too much to expect that the development of the destroyer will continue at the same rapid pace that it has set during the past five years of war.—"International Marine Engineering."

TRAINING OF ENLISTED PERSONNEL OF THE UNITED STATES NAVY FOR ENGINEERING DUTIES.

By Lieutenant Commander R. R. Smith, U. S. N.

At the outbreak of the war the Bureau of Navigation of the Navy Department was confronted with the problem of supplying trained engineer personnel to man not only the vessels of the Navy in reserve, which were then being rushed into active commission, but the interned German and Austrian ships which were completing the repairs necessitated by the sabotage they suffered during their internment. New construction in excess of anything heretofore attempted was being planned, while construction under way was being rushed to completion. Deck personnel was required also, but, as Navy training has always chiefly concerned itself with the training of its deck personnel, facilities and methods were already available for this training and only required expansion. The training of the engineering personnel has, however, always been a slower process on account of the greater degree of skill required in the handling of mechanisms. Our merchant marine was small and the comparatively few merchant engineers could not be spared to the Navy. Naval recruits by the thousands could be sent to these ships, but the sending of green men to do the work of trained engineers and firemen was rejected as impracticable.

A conference was called at the Bureau of Navigation, which was attended by representatives of the commander-in-chief of the Atlantic fleet. The representatives from the fleet proposed a scheme which meant a new departure for the Navy. It required setting aside a great many old ideas, but it sounded good. The conference adjourned until the next day, when the details which had been worked out over night were presented. They were accepted, and at the suggestion of the chief of bureau it was decided that an officer should be ordered to the staff of the commander-in-chief to follow up the development of the scheme proposed.

THE PROPOSED SCHEME.

The salient features of the proposed scheme as outlined in the commander-in-chief's letter were as follows:

(1) Eight battleships of the older type to be assigned to the paramount duty of training engineers.

(2) Reduction of deck complement to one-half to make room for the increased number of engineers.

(3) Greatly increasing the engineer petty officers to provide suitable instructors.

(4) Removal of practically all firemen from ships assigned to this duty.

(5) Increasing the number of trained engineer officers for each engineering ship.

The only two ships available at the time were the *Alabama* and the *Kearsarge*. Later six more ships joined the training squadron. These ships were ordered to revise their complements immediately and to transfer all engineer personnel except ninety men. Additional petty officers to make up the ninety in the required ratings were transferred from battleships not engaged in this work. All battleships were to regard one-third of their engineer complement as available for transfer, being obliged to train recruits to this number for their own vacancies.

A senior assistant to the engineer officer and two commissioned officers of engineering experience for training officers were ordered to each training ship. The complement of petty officers was as follows:

Chief machinist mates	9
Chief water tenders	9
Machinist mates, first class	9
Water tenders	27
Oilers	18
Firemen, first class	18
Total	90

The firemen were for the ship's use on evaporators, ice machines, steamers and other auxiliaries where some permanence was essential. Each ship was required to train the following men at one time:

Engine drivers	16
Oilers	32
Water tenders	32
Firemen	200
Total	280

There was thus a total of 370 engineers on board, not including the electrical force, which brought the total to nearly 400. The regular engineer complement of the *Alabama* for war conditions was 170, and during peace much less than this. In order to facilitate the work of receiving, stationing and transferring so large a number of men, it was decided that half be transferred at a time. Six weeks was decided upon as the training period. In order to alternate the transfer and commence the system, half of the men had to be transferred at the end of the first three weeks. The better half were selected, this being possible owing to the fact that the first men received came from the other ships, had been on board anywhere from one to three months and had got some training before being received on board.

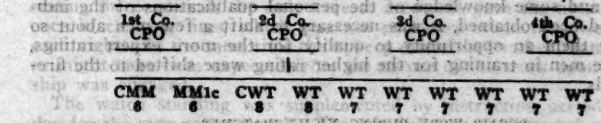
Upon the internal organization of the ship depended the success of the entire scheme. It was realized that the moment the individual man under training did not receive close personal supervision by some one person, the entire scheme would fall down. The entire organization was based on this idea and the squad system was inaugurated.

It has been stated that no one person can intelligently supervise the entire work of more than eight men; when the number is increased beyond this the closeness of the supervision is relaxed.

SQUAD SYSTEM INAUGURATED.

The men under training were divided into four companies. Each officer had two companies, each company was commanded by a chief petty officer. The company consisted of seventy men under training, divided into ten squads, each squad being commanded by a petty officer not below the rating of first class. Two of these squads were engine room squads, of six men each, with a machinist mate in charge of each. One squad was in training for water tenders and was in charge of a chief water tender; seven squads were in training for firemen and each squad was in charge of a water tender.

Diagrammatically expressed, the organization was made up as follows:



Each engine room squad of six men was made up of two men in training for engine drivers and four men in training for oilers. The ship's oilers were utilized as instructors on watch, but not as squad leaders. In conducting the training the squad system was adhered to throughout.

DEPOT DIVISIONS ESTABLISHED.

When the scheme was first put into operation each ship was assigned a Depot Division. Each battleship division in squadrons three and four were assigned two training ships for the purpose of transfer and receipt of men. The trained men were sent back to the Depot Division ships, which in turn transferred them where needed. A steady inflow of recruits was supplied to the Depot Ships, to supply the needs of the training ships.

Recruits without previous experience were sent for training as firemen, firemen of naval experience were sent to be trained as water tenders, while recruits having mechanical experience in civil life were sent for training as oilers and engine drivers. The latter term was devised to supply men who could operate machinery. It was thought that men could be made available for the specific duty of handling an engine much more quickly, if their training was confined to this particular subject. It was realized that any attempt to train men for general duties in the engine room would require a great deal more time than was available. The need for men was urgent and pressing.

RECRUITS OF MARKED INTELLIGENCE.

Much of the success of the training was directly attributed to the fact that the recruits were men of an unusually high order of intelligence, nearly all of them having enlisted for voluntary service upon the outbreak of war. Many excellent mechanics were enlisted under the rating of machinist mate, second class, and, while they had no marine experience, they made excellent progress. Among the firemen were found many men who could have been enlisted as machinist's mates. Most of them, however, had rushed into the service in any rating. No difficulty was experienced in obtaining men for the engine room branches, but it was soon found that the supply of firemen of experience for training as water tenders was running low. A water tender can learn his regular duties without much difficulty, but experience in fireroom mishaps is essential as an element of safety. In training new men for this rating it was decided that they would not be rated, but a notation made in their records that they had been trained for this work.

The men showed the greatest enthusiasm for their work. They realized that everything possible was being done to instruct them in their future duties; that they would soon be cast upon their own responsibilities and would be expected to produce results. Men who received a certain mark were recommended for one higher rating; in some cases men of unusual ability were jumped from third class firemen to oiler, machinist mate second class or firemen first class. After a new draft had been on board one week and some knowledge of the personal qualifications of the individuals had been obtained, it was necessary to shift a few men about so as to give them an opportunity to qualify for the more expert ratings, while some men in training for the higher rating were shifted to the firemen squads.

REPAIR WORK DURING NIGHT WATCHES.

During the first months of training the ships got under way at dawn and steamed until sunset, banking fires over night. On the *Alabama* all repair work was undertaken during the night watches. Feed pumps were opened, condenser tubes repacked, bearings adjusted, and everything closed up before four A. M., when steam was turned on the main steam line for warming up. We welcomed the opportunity to repair something and required the men under training to do all the work under the supervision of the petty officers. It is surprising how quickly a man can learn to do something if he is allowed to do it. While under way all the firing was done by the men under training under the supervision of the squad leader. The water was tended by the men under instruction for that rating, under the supervision of the squad leader, a chief water tender. The engines were handled and oiled by new men. It was difficult to persuade the old men that they must allow the recruits to do everything, but they finally got the idea, and later rather enjoyed it.

On a Monday morning after the receipt of a new draft the ship would be got under way by an entire crew of green men. Sometimes, upon getting under way, a signal of one-third speed ahead, followed by another of two-thirds ahead, and another of standard speed would be received from the bridge before the student could get the engine turned over. But no one was allowed to interfere, and unless the situation on the bridge was known to be urgent they were allowed to wait and suffer the delay. It must be said that the commanding officers made the best of the situation, and, while regretting the subordination of the military mission of their ships, lent their heartiest co-operation to the engineer officers.

As the interned ships were completing their repairs, the Depot Ships were discontinued and the men sent directly to the receiving ships. Each draft was detailed specifically for the ship to which it was going and the interest ran high. It was necessary in self-defense to obtain and post all data available on the ships in question to satisfy the curiosity of the men. By so doing it lent interest to the training. When the *Vaterland*, now the *Leviathan*, was commissioned, each man wanted to be sent to her. When there was no objection, each of the top men in their squads were allowed to ask for the ship they wanted, and their request was granted, if possible. The men in the deck force commenced going to the mast, requesting to be put in the training draft. Some of these requests were granted.

MEN SENT TO CAMPS AT NAVY YARDS.

After the commissioning of the interned ships, the men who completed the training course were sent to the camps at the various navy yards. The

new drafts were received directly from the receiving ships and this method was steadily followed.

As stated above, the training ships operated during the day only during the first months of training. Later, however, all ships operated with the fleet, and the training afforded the men below, while not so good for the engine drivers, was better for the firemen, oilers and water tenders, inasmuch as the ship was operating under conditions more closely approximating the conditions the men were to meet. It gave the firemen experience in cleaning fires, which was not necessary when fires were banked each night. At the same time the upkeep diminished rather than increased, as might have been expected from the nature of the work upon which the ship was engaged.

The watch standing was supplemented by instruction periods, twice a day for the men not on watch. The schedule was so arranged with regard to the watches that all men in training for the same rate received the same instruction.

The following courses of instruction were given. These were supplemented by lectures on naval usages and customs of the service:

COURSE OF INSTRUCTION.

ENGINE DRIVERS AND OILERS.

Feed Water Supply.

(1) Height of water—methods of ascertaining.
(2) Supply of feed water—how obtained.
(3) Action necessary at failure of feed water.

Main Engines.

(1) Throttle valve, butterfly valve, by-passes, bleeders.
(2) Reversing engines, link gear and cut-offs.
(3) Cylinder drains and relief valves.
(4) Gage board.
(5) Bearing, main, thrust, and steady. Sterntube.
(6) Piston rods, valve stems, crossheads, gibs and guides.
(7) Engine and fireroom telegraphs and counters.
(8) Vacuum and loss of same.
(9) Oil—grades used for different purposes.
(10) Oil service and water service.

Main Engines and Auxiliaries.

(1) Feed pumps, main and auxiliary.
(2) Air pumps, circulating pumps and condensers.
(3) Feed heaters, separators and traps.

Feed and Steam Lines.

(1) Feed tanks, reserve tanks, pumps, connections, feed lines, main and auxiliary.

Ship's Drainage, Fire Main and Flushing System.

WATER TENDERS AND FIREMEN.

Boilers and Firerooms.

(1) Coal tally and passing coal.
(2) Priming, lighting off, steaming level.
(3) Firing.
(4) Cleaning fires and ejecting ashes.
(5) Care of bilges and strainers.
(6) Feed water—height of, supply, how obtained, action necessary for failure of water.

(7) Bunker gases—safety precautions.
(8) Time firing device.

Feed Lines.

(1) Feed pumps, main and auxiliary.
(2) Feed heaters, separators and traps.
(3) Feed tanks, reserve tanks, pumps and connections.

Ship's Drainage and Flushing System—All Boiler Connections and Fittings.

(1) Safety valves, feed checks and stops, bulkhead stops.
(2) Steam traps and cut-outs.
(3) Water columns and try cocks.
(4) Drums, handholes, manholes, door casings.

Each subject was developed by a questionnaire for the use of the squad leaders, and there is in use on board at the present time three pamphlets, one for each course. An example of one of these lessons is given below:

COURSE FOR ENGINE DRIVERS AND OILERS.

Lesson No. 12.

Subject.	*Reference.*
1. Feed Water System.	Ship's Installation. Eng. Handy Book, pp. 85, 155, 158 and 161.

Questions may be made up for this lesson from references given.

Q. —Describe the following parts of the feed water system, including the material of which they are made.

How do they operate?
How are they connected to other parts?
How are they cleaned?
What care and preservation is required?
What troubles are encountered?

1. Reserve feed tanks.
2. Feed tanks.
3. Feed tank filters.
4. Feed pumps.
5. Feed heaters.
6. Grease extractors.
7. Feed lines, main and auxiliary; expansion bends.

Q. What is back pressure? Where is it used? How much is carried? How is it regulated? What is a hot well? Where is it? Why? What is in it? Why? How often are filters cleaned? Trace the main feed line from the feed tank to the boilers, naming all the valves, pumps, expansion bends, connections and cross-connections and other auxiliaries in the line. Why are there two feed systems? Tell where they are and what cross-connections are between these two lines. When would you cross-connect them? How would you put feed water into the reserve tanks? How would you get it out?

The course was developed on the *Alabama* into the series of lessons of which the above is an example. They were made as simple as possible in order that all might understand them. The course as published is in use on all the training ships. Each ship has developed the subjects, while several have adopted the *Alabama* pamphlets.

At the conclusion of the course a page was inserted in the enlisted record of each man, giving the following information:

INTENSIVE TRAINING—ENGINEERS.

(C-in-C Letter No. 1885 of 4-24-17.)

Trained on board U. S. S. .....
Name .....
Actual rate .....
On transfer from U. S. S. .....

Instructed as .....
General proficiency .....
Qualified as .....
Type of boiler .....
Type of engine .....
Lubrication of engines .....
Fuel .....
Received from U. S. S. .....
Date received .....
Transferred to .....
Date transferred .....

The training has been in operation for about fifteen months at the time of writing this article. Approximately 16,000 men were trained in this time. These men are now doing duty on all types of ships, including battleships, cruisers, transports, destroyers, yachts and submarines. On a recent trip abroad the writer had an opportunity of questioning a great many of the men who had passed through the training course. Many of them had been advanced in rating. This is the best indication that their training had been satisfactory. The reports received from the officers who were working with these men were very satisfactory.

What was a great problem at the beginning of the war has been solved in a very simple manner. If it had not been satisfactorily solved, it is believed that the Navy would have been seriously hampered in its operations. The absence of fireroom and engine room casualties must indicate that these men who have come into the Navy since the commencement of the war have been making good. The officers who have had a part in this work can feel that they have contributed directly and effectively towards the excellent showing of the Navy during the past year.—"International Marine Engineering."

THE PROPULSION OF SINGLE-SCREW SHIPS.

The determination of the horsepower necessary to drive any particular ship at a given speed may be made by the choice of a suitable Admiralty constant or by model experiments. In the former all the factors which comprise the performance of the ship are included, and no special calculations are required to ascertain the resistance of the ship or the efficiency of the screw. Certain advantages therefore are to be gained from using the constant methods, but these are nullified to some extent by the uncertainty as to the constant to choose. Care must be taken to fix one which from previous actual experience has been obtained in a vessel very close to the type being powered. When the model experiment method is adopted the problem is only partly solved, as the effective horsepower thus obtained forms, roughly only one-half of the total power to be installed. The choice of the proper propulsive efficiency thus becomes an important item, and this can only be done with satisfaction when the effective and indicated or shaft horsepowers of a previous vessel of similar type are known.

In the analysis of the performances of ships it has been a matter of some surprise to find that in spite of the high hull efficiency single-screw propulsion does not appear to be much superior to that with twin, triple or quadruple screws. When Sir William White many years ago gave a paper to the Institution of Naval Architects on "The Speed Trials of Recent Ships of War," this feature was referred to. Many times since then has the fact that single-screw ships do not surpass twin-screw vessels in gross propulsive efficiency been alluded to. In the case of the *Vespasian*, for example, the results of which were given by Sir Charles Parsons to the Institution of Naval Architects in 1910, the propulsive efficiency was only 53 per cent with reciprocating machinery, a performance which has often been excelled by twin-screw vessels.

It has been suggested many times that the efficiency of a good propeller should be in the region of 70 per cent. If this figure were accepted, the difficulty of accounting for the moderate propulsive efficiencies of single-screw ships becomes the more acute. Now it is probable that the majority of screws in common use may have such an efficiency when revolving at some particular slip, but it is certain that, when revolving, such screws are exerting the necessary thrust to drive the vessels to which they are attached at their particular speed, the real slip at which this thrust is given is greatly beyond the point of maximum efficiency. This point has been admirably illustrated by the latest experiments of McEntee in the Washington tank.

A certain propeller was attached to four separate models and propelled in the Washington basin. It was found, by independent experiment, that this particular screw had a maximum efficiency of 71 per cent at about 13 per cent real slip. The efficiency curve then fell rapidly, until at 53 per cent real slip the efficiency was only 50 per cent. The models were of a full cargo ship and the slips at which the screw was working when the necessary thrust was given varied from 40 to 50 per cent in the four models tried. The actual efficiency of the propeller within these limits was 60 per cent. On a basis of shaft horsepower the hull efficiency value was 1.05, the wake being greater than the thrust deduction loss. If it be assumed that the mechanical efficiency of reciprocating machinery is 90 per cent, this 63 per cent on the shaft horsepower basis would be reduced to about 56 per cent on a basis of indicated horsepower—a not unreasonable figure.

It must be remembered that, although in single-screw vessels the propeller is placed most advantageously from the wake standpoint, a large thrust deduction factor is generally associated with these high wakes.

In the measurement of wake values in a model basin it is possible that some errors arise, not necessarily in the determination of these factors, but in their direct application to full-sized ships. In model experiments the frictional resistance is a greater proportion of the total than is the frictional resistance of a full-sized ship; that is, a correction must be applied to the model resistance in order to arrive at the proper ship resistance. If, then, the frictional resistance is magnified in model results, is it not more than likely that the frictional wake is also greater than would obtain in the actual vessel? What the difference would be is problematical, but it undoubtedly would have the effect of reducing the hull efficiency as measured by tank methods and go in the direction of accounting for the obtained propulsive efficiencies in single-screw ships.

The main point, however, is in the loss in efficiency due to the great slip at which the majority of single screws operate. When limited diameters obtain, the thrust necessary to propel a full slow form can only be produced when the screw is revolving with great slip, and this means on the descending point of the efficiency curve.

It will be interesting to learn from the results of the trials of the numerous cargo ships being fitted with geared turbines if the hopes of obtaining relatively high propulsive efficiencies with these are obtained. Unfortunately, the war retarded the adoption of this type of machinery in cargo ships, and insufficient data are at hand regarding the sea performances of geared turbine vessels. Should the hopes be realized, a great step will have been made towards the attainment of higher efficiencies and great economies.—"International Marine Engineering."

INTERNAL COMBUSTION ENGINED WARSHIPS.

Lord Fisher's recent communications to the "Times," urging economy and suggesting characteristically sweeping means to that end, have raised afresh some controversial points. The nation is not unmindful of the contribution in the past of Lord Fisher in adopting strong measures to ensure at all costs that the efficiency of the Fleet should be maintained at the highest possible value. A reminder is hardly needed of the part played in securing such efficiency of fighting force by the water-tube boiler, the steam turbine, the big gun, and the submarine, or of our indebtedness to Lord Fisher in his advocacy of these important factors. As a result, such improvements were effected in naval engineering that when the war commenced we led the world by a good margin. Lord Fisher also realized at an early date the potentialities of the internal-combustion engine as a means of propulsion for fighting ships.

This subject was first raised in a paper before the Institution of Naval Architects given by Sir James McKechnie on March 20, 1907. Later on, it will be remembered, a Royal Commission was set up at the instigation of Lord Fisher, and presided over by him. The findings of this Commission have not been made public, but it was generally recognized that the report was a disappointment to Lord Fisher to the extent that it indicated that at the time when the inquiry was held a great amount of development still remained to take place before the internal-combustion engine could be regarded as assailing the first place in naval propulsion firmly and justly occupied by the steam turbine. The Commission certainly served a useful purpose by the collection of evidence and data which showed clearly the state of the internal-combustion engine at that time and indicated the work to be undertaken and the problems to be solved before a reconsideration of the application of the Diesel engine to naval work of high power need arise. Oil having been proved in practice in the interval to be the fuel *par excellence* for naval use, and the Diesel being that type of internal-combustion engine best suited to utilize that fuel with maximum economy at relatively high powers (for this class of prime mover) our present remarks may be confined solely thereto.

Lord Fisher's recent letters to the press have given rise in the minds of many engineers to that reconsideration. During the war experiments on such subjects would not have been justified, and reliance was placed rather on combinations of well-tried units, changes being made only after full consideration of the maximum possible detrimental effect on fighting efficiency should the anticipated improvement be completely negatived. Such, however, was the mass of the production of naval machinery undertaken during the war that short steps carried us relatively long distances, and in effect the progress made during this period in most of the branches of naval engineering was greater than in any previous period of time of equal length. The one exception is undoubtedly the Diesel engine. The same 100 brake horsepower per cylinder that was standard in our sub-

marines before the war was the standard, except for minor improvements, at the end. The Germans in their submarines had engines of some 300 brake horsepower per cylinder (see "Engineering," "German Submarine Diesel Engines," June 13 last, page 763), and the Clyde during the war produced a merchantman with two main engines of 16 Diesel cylinders, developping in normal running at sea a total of 6,600 indicated horsepower, or over 330 brake horsepower per cylinder. The engines referred to are the four-cycle single-acting Diesel engines of the motor ship *Glenapp*. The cylinders have a diameter of 760 mm. (29.92 inches) with a stroke of 1,100 mm. (43.31 inches) and run at 125 revolutions per minute. These are not experimental engines, although much work of such a nature has been undertaken and remains to be recorded. Before the war many rumors concerning Diesel engines building on the Continent for battleships were current. The facts are briefly as follow: The M.A.N. Company, at Nuremberg, built a three-cylinder two-cycle double-acting engine, from which several thousands of horsepower were expected. The results, however, were disappointing, and a serious accident—an explosion in the scavenging system setting fire to the staging and screens erected round the engine for secrecy—which, however, concerned only very indirectly the main principles of the Diesel engine, had the direct effect on the Continent of stopping the great amount of experimental work of very similar nature in process at that time. Messrs. Krupp in Germany built a 2,000 brake horsepower double-acting two-cycle cylinder, with which no substantial success was achieved. In France Messrs. Schneider, of Le Creusot, under license from Messrs. Carels, of Ghent, constructed a 1,000 brake horsepower single-acting two-cycle engine, regarding which no detailed results are available.

Perhaps the most successful of such units was that of Messrs. Sulzer, of Winterthur, where a single-acting two-cycle cylinder of 39.4 inches diameter by 43.3 inches stroke developed at 150 revolutions per minute, it is claimed, 2,000 brake horsepower, with a fair measure of success, sufficient, at any rate, to justify the builders in standardizing a two-cycle cylinder of 600 brake horsepower with a cylinder diameter of 29.5 inches and a stroke of 39.4 inches, running at 130 revolutions per minute. A six-cylinder engine of these units gives 3,600 brake horsepower, and of these several have been built for land work.

These pre-war examples indicate the progress made up to that date, and it must be admitted that the large experimental engines were generally unsuccessful, even although they were designed and built with the one and only aim of giving the maximum horsepower regardless of any other considerations such as weight, compactness, piston speed or speed of revolution. The explanation is to be found in that they were too far ahead of the knowledge at that time of this type of engine. In some cases the experiment may have been justified on the score that the information gained was applicable to and effected improvement with engines of moderate power. The *raison d'etre* of this pre-war campaign was that Germany, in many ways the home of the Diesel engine, had lagged behind with the steam turbine, but hoped by intensive development, government fostered, to regain, with this prime mover, the place she had lost on account of our progress meantime with the steam turbine. In this Germany failed, although her engine builders can claim that their largest war submarine engines showed latterly a power capacity per cylinder of nearly three times that achieved by the British standard engine.

Another landmark is the 400-500 brake horsepower submarine unit cylinder of 350 revolutions per minute, designed and built by Messrs. Sulzer during the war, with which highly satisfactory results have, it is

believed, been obtained. Six-cylinder engines composed of these units have been built developing 3,900 brake horsepower per engine.

These examples indicate in a general way the various steps in the development of the Diesel engine up to the present time. The question then is, can we assess a value to this progress in comparison with the total distance required to be travelled before such prime movers can be considered for first-line fighting units of the fleet? In view of the success of the steam turbine such a consideration will be regarded by some as unnecessary.

The Diesel engine possesses potentialities for fighting ships that demand repeated attention in order that they may be equated to naval requirements if and when the known deficiencies are remedied. The attractions of the Diesel engine are: Its oil economy, using the most suitable naval fuel, far surpassing the best results achieved by or predicted for the steam turbine; its readiness for immediate service without requiring warming through, etc., the relative ease with which exhaust gases can be discharged compared with these for steam plant, the better arrangements of guns and machinery due to the elimination of the uptakes and the substitution of a relatively small exhaust pipe for funnels. Against these advantages must be set against the disadvantages of small power per unit cylinder, the high weight, height and uncertainty of operation, especially with large units, under certain practical conditions.

Instancing three standard powers of naval vessels, the small battle cruiser and the battleship, and light cruiser and torpedo-boat destroyer, the first of 80,000 shaft horsepower, and the others of from 40,000 to 27,000 shaft horsepower, and assuming cylinders of 1,000 brake horsepower, 80, 40 and 27 would be required. The maximum number of Diesel cylinders in any vessel afloat is found in the "J" class submarine boats, with their 36 cylinders (three 12-cylinder engines). The maximum number of units which can be regarded as suitable is therefore relatively large. Twelve turbines and more than twelve boilers are not regarded as excessive in steam practice, and 36 cylinders in the case of internal combustion engines have, as already mentioned, proved a sound arrangement.

In respect of increase of power per unit with Diesel engines, progress, it will be seen, has been relatively slow. Up to 300-400 brake horsepower can be said to be proved at the present time to be practical politics. Higher powers still can no doubt be made to give quite satisfactory service, but the demand hitherto has been severely limited by the remarkably rapid success of the application of geared turbine machinery for vessels of all powers, down even to relatively small installations. For naval work, however, the advantages of internal combustion create an insistent demand.

Success in the past with increasing power of Diesel engines has not been achieved without encountering many difficulties. Chiefly, it must be stated, on account of insufficient attention being paid to small and intricate but absolutely essential details, of which with this type of engine there are so many. Knowledge is increasing, and a satisfactory cylinder developing 1,000 brake horsepower may confidently be anticipated in the near future. Whether, however, it will meet naval requirements or not is another matter, depending essentially on considerations of weight, space occupied, and such factors. We discussed the future of this type of prime mover in regard to these terms in an article entitled "Internal-Combustion Engines for Submarines and Aircraft,"—published in "Engineering," of September 20, 1918, page 319—and held that the future would see in such work much progress, as has been the case with other prime movers, notably the petrol aero engine. The internal combustion turbine also has been discussed, but still remains no more than a fervent hope.

It is too early yet to speak in any definite way of future naval policy, yet, in all probability, the efforts of the next few years, at any rate, will be concentrated upon the development of those types of fighting units that have proved themselves or have indicated the path of progress. This policy will entail considerable experiment, in which the Diesel engine will not be overlooked. Such work is always costly, and whether or not the results are dearly bought depends largely upon the method of control, the imagination brought to bear, and the careful consolidation of each step gained.—"Engineering."

DEVELOPMENT OF GEARED TURBINES FOR THE PROPULSION OF SHIPS.

By Mr. R. J. Walker.

Although the successful application of the steam turbine to marine propulsion dates back to the year 1897, when Sir Charles Parsons demonstrated in that now historical vessel, the *Turbinia*, the great advantages of the turbine system when applied to the propulsion of ships, it is only within the last few years that mechanical reduction gearing has been largely adopted in association with steam turbines.

Sir Charles Parsons always had in mind the possibility of reduction gearing in connection with the application of his steam turbine to marine propulsion, and even as far back as 1897 a small launch was constructed in which helical gearing similar to that adopted by Dr. De Laval, of Stockholm, with his high-speed turbines for land purposes, was introduced between the turbine and the propeller. This is believed to be the first application of helical gearing to drive a propeller.

It was considered desirable, in view of the very wide field which was opened up by the success of the *Turbinia* for the application of direct-driven turbines to high-speed vessels, to confine attention for a time to development in this direction only, and to defer embarking on the additional pioneer experimental work incidental to reduction gears of unprecedented size. The wisdom of such a policy was subsequently shown by the great difficulty experienced at the commencement in introducing the turbine system of propulsion against the conservatism of British engineers and shipowners who looked upon such a revolution in the methods of marine propulsion with a great amount of doubt and distrust. Very little experience existed at that time of the accurate cutting of toothed gearing in large sizes, and the additional work involved would certainly have added to these difficulties.

Before proceeding to consider the development of geared turbines, it may be of interest to briefly review the progress made in connection with direct-driven turbines.

Following on the success of the *Turbinia*, the torpedo-boat destroyers *Viper* and *Cobra* were built and fitted with turbines for the Admiralty in 1899, and achieved remarkable speeds.

The next step in the case of war vessels was when the system was fitted to the third-class cruiser *Amethyst*, three other sister vessels being fitted at the same time with reciprocating engines. The results of these trials demonstrated the economy of the turbine in this class of vessel, more especially at the higher powers, and influenced the British Admiralty in deciding to have the battleship *Dreadnought* fitted with turbine engines. The keel of this vessel was laid down at Portsmouth in October, 1905. By 1906 the British Admiralty had adopted turbines for all new construction.

In the early endeavors to introduce the turbine for commercial purposes,

no shipowner could be found bold enough to make the experiment on any reasonable terms. It was ultimately recognized that the only chance of convincing shipowners of the practicability of such a proposal was to build a vessel specially for the purpose, and Captain John Williamson, Messrs. Denny Brothers, and the Parsons Marine Steam Turbine Company, Limited, undertook the financial responsibility of building the Clyde steamer *King Edward*, which was first put into service in the summer of 1901, and has been running satisfactorily ever since that date. Other vessels very quickly followed the *King Edward*, until at the present time the total shaft horsepower of turbine vessels on service for commercial purposes is about 1,600,000.

One of the chief difficulties which had to be met in applying the steam turbine to the propulsion of ships arose with the propellers. As is now generally known, it is desirable that a turbine for maximum efficiency should run at a high rate of revolution, whilst on the other hand, for maximum propeller efficiency, much lower rates of revolution are necessary. Obviously one solution was by the introduction of some form of gearing between the turbine and the propeller; the time, however, for such a departure, for reasons already mentioned, had not then arrived. The problem was therefore to reconcile as far as possible those two opposing factors, and to arrive at the best compromise to meet the conditions required by a suitable lowering of the revolutions as then designed for land turbines, and a raising of the revolutions of the propeller by suitable modifications in the form. This was one of the most serious problems which had to be faced in the design and construction of the *Turbinia*, and led to many modifications in the propellers and turbines of that vessel before success was finally achieved.

With the raising of the revolutions of the propeller the diameter and pitch ratios were less, whilst the ratio of blade surface to disc area was much greater than is usual with propellers driven by ordinary reciprocating engines.

It was found as a result of numerous and costly experiments that a judicious compromise could be, and was, arrived at, whereby satisfactory combined efficiencies of turbines and propeller were obtained for high-speed vessels, resulting in increased speed of ship and increased economy in coal consumption.

The chief governing factors in marine steam turbine designs are those of economy, weight, and first cost, and it was found in actual practice that the problem of applying the turbine direct to the propeller was for the time satisfactorily solved to fulfill these conditions for vessels of about 18 knots speed and upwards.

Up to the year 1909 the steam turbine had not been applied to vessels of slow and intermediate speeds, with the exception in a few instances of the combination of reciprocating engines with a low-pressure turbine.

From the early years of steam turbine design, it became apparent that the turbine engine was capable of economically dealing with ratios of expansion beyond the limits possible with reciprocating engines, and a combination of reciprocating engines with a low-pressure turbine presented advantages as regards economy in that class of vessel where the designed full speed fell below the range suitable for an all-turbine arrangement, that is to say, for vessels of about 18 knots speed and under, the reciprocating engines in such an arrangement working in the region of pressure drop where the conditions are best suited to it, and the turbine utilizing that portion of the expansion diagram which the reciprocating engine is not able to utilize efficiently. The service results of vessels fitted with this combination system have shown that this arrangement of machinery

effected a saving of from 12 per cent to 14 per cent in coal consumption as compared with similar vessels fitted with quadruple engines.

It was not until the success of the marine steam turbine had been assured by its rapid adoption in vessels of moderate and high speed, and more time was available, that attention was directed to the application of the turbine to slow speed vessels by the introduction of gearing between the turbine and propeller. Several forms of gearing have been proposed, such as electrical, hydraulic, and mechanical. Electrical and hydraulic transmission gear have been fitted in a few ships, but the greater majority of vessels have been fitted with mechanical gearing.

In view of the success obtained by Dr. De Laval, of Stockholm, with helical and double helical gear in connection with his turbine for land purposes for powers up to about 600 brake horsepower, Sir Charles Parsons decided to test turbines mechanically geared to the screw shaft. Even at this stage there was encountered a very strong prejudice to gearing, and it was found necessary, in order to test its practicability for marine work, to carry out experiments in a cargo boat.

In 1901 an old steamer, the *Vespasian*, was purchased and fitted with geared turbines in place of her original triple-expansion reciprocating engines, but before the change was made exhaustive trials were carried out with the original engines. Full particulars of the results of the comparative trials of the two systems of machinery were given in a paper by Sir Charles Parsons read before the Institution of Naval Architects in 1910. It will be sufficient here to state that an additional economy in coal consumption of over 15 per cent was obtained in the *Vespasian* by the substitution of geared turbines for reciprocating engines, and that the efficiency of the mechanical gearing was found to be about 98½ per cent. The *Vespasian* was run between the Tyne and Rotterdam, carrying coals, and in general trade for four years, and the hull being then worn out was broken up, but the turbines and gearing, which showed no signs of deterioration, were taken out of the vessel and fitted to a new steamer, the *Lord Byron*.

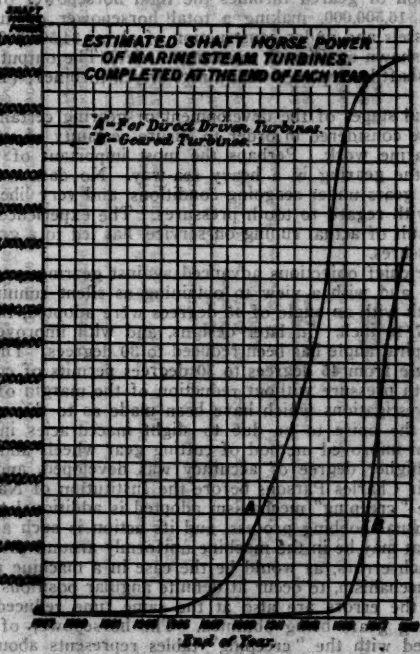
The success of the *Vespasian* aroused very keen interest amongst engineers and shipowners, and Professor Sir John Biles was amongst the first to recognize the advantages of increased efficiency to be derived from geared turbines for cross-Channel steamers when he recommended the London and Southwestern Railway Company to adopt geared turbines in their new steamers *Normannia* and *Hantonia*, for their Southampton to Havre service, and built by Messrs. the Fairfield Shipbuilding and Engineering Company, Limited. The trials of those vessels, which took place in the early part of February, 1912, were successful, and the ships have continued to give very satisfactory results.

The *Paris*, another cross-Channel steamer for the London, Brighton and South Coast Railway Company, was built a year later by Messrs. Denny and Brothers, and fitted with geared turbines, and attained a speed of 24¼ knots. After the success of these vessels, other shipowners quickly followed the lead of these companies.

Although gearing was primarily introduced to widen the field of operation by its adoption for vessels of low speed, it was soon recognized that increased efficiency could be obtained by means of reduction gearing in fast ships. The Admiralty placed an order in 1910 for the part gearing of the machinery of H.M. torpedo-boat destroyers *Badger* and *Beaver*, by introducing gearing for the high-pressure and cruising turbines of these vessels. In 1912 the Admiralty adopted gearing for the whole of the machinery of the two destroyers H.M.S. *Leonidas* and *Lucifer*, of 22,500 horsepower, fitted on two shafts.

Exhaustive trials of these vessels were carried out, the results of which showed a considerable increase in efficiency both at full and cruising speeds as compared with previous torpedo-boat destroyers with direct-driven turbines.

The adoption of all-geared turbines on two shafts in these destroyers permitted an increase in the propeller efficiency of about 12 per cent, and an additional improvement in the steam consumption of the turbines, of about 10 per cent at full power, and about 30 per cent at one-tenth of full power, also a slight saving in the total weight of the machinery, as compared with a twin screw arrangement of direct-driving turbines hitherto adopted in these vessels.



During the war gearing was universally adopted for light cruisers, battleships and battle cruisers of the highest powers, and during the last four years practically no direct-driven turbines have been fitted. Vessels are now in service with installations of geared turbines of 100,000 horsepower, and the horsepower transmitted through a single gear wheel has reached 25,000, and the power through a single pinion 15,500.

The further adoption of geared turbines in the mercantile marine was considerably retarded during the war. When the question of type of machinery came under discussion for the standard ships, preference was

given to reciprocating engines owing to the fact that all the usual builders of marine turbines were full of orders, and were required to give preference to the manufacture of turbines for war vessels, which limited the number of turbine engines which could be built for commercial vessels, and also to the fact that the country had great resources at its disposal for the quick manufacture of ordinary triple-expansion engines. In the later standard ships of the fabricated type, however, geared turbines were adopted.

At the present time the total horsepower of geared turbines for vessels for commercial purposes completed and under construction is about 1,400,000; and the total horsepower of geared turbines for war and commercial purposes completed and under construction is about 18,000,000. Previous to the adoption of geared turbines the total horsepower of direct-driven turbines was 16,500,000, making a total horsepower of marine turbines fitted into vessels and under construction of about 34,500,000.

The curve shown on Plate I represents: (A) The output at the end of each year in shaft horsepower with direct-driven turbines; (B) with geared turbines.

In the initial stages of the development of gearing certain factors were required to be considered in order to ensure certainty of success with the gears for marine work. Perhaps the most important of these was the behavior of the gearing in a heavy sea-way. No data was available as to the life of gears under seagoing conditions, and very liberal allowances were made with regard to tooth pressures. The experience gained, however, as a result of actual runing on service, has led to a gradual increase of tooth pressures.

One of the chief objections advanced against gearing was the anticipation of noise and with a view to obtaining as silent running as possible fine pitch teeth with an angle of 45 degrees were adopted, the pitch of the tooth being 7/12 inch. In later designs, and with improved methods of cutting, the spiral angle has been reduced to 30 degrees. The reduction of the spiral angle from 45 degrees to 30 degrees permits of considerable increase of tooth pressure without reduction of the margin of safety.

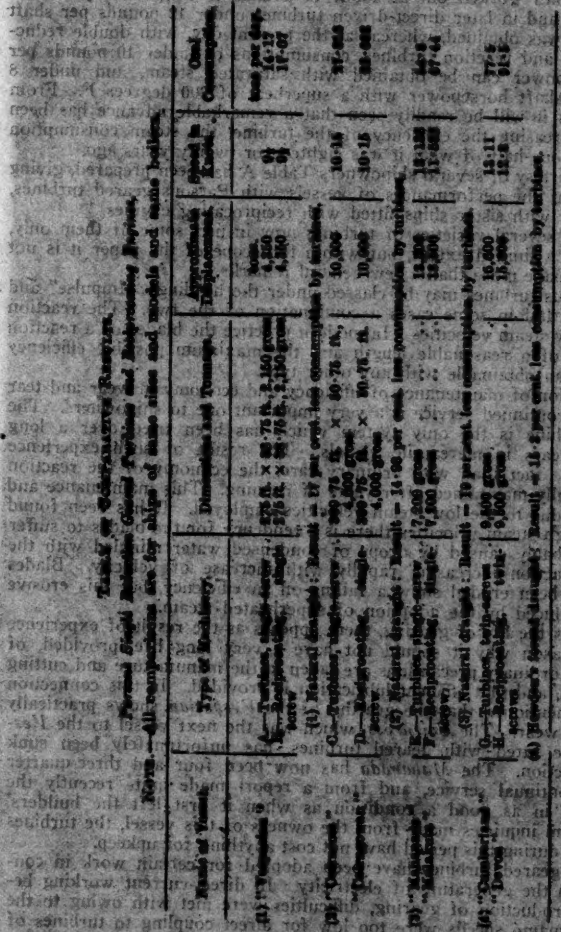
Careful investigations which have been made as to the causes of noise have shown the noise to be due to slight inaccuracies in the pitch of the teeth. An improved method of cutting gear wheels and pinions so as to obtain a higher degree of accuracy was developed, and described in a paper by Sir Charles Parsons before the Institution of Naval Architects in 1913. The "creeping" mechanism adopted is additional to the machine used in the usual hobbing process, and its action is such as to distribute any errors that may be in the machine uniformly around the wheel instead of leaving such errors, as would be the case in a machine not fitted with "creeping" mechanism, to occur at definite angular positions on the wheel. Incidentally, the errors are also at the same time reduced, resulting in quieter running gears being obtained. The horsepower of gears cut by machines fitted with the "creeping" tables represents about 75 per cent of the total in use.

Within the last three or four years a further development has taken place in the case of mercantile vessels by the adoption of double-reduction gearing instead of single-reduction gearing, which had previously been the practice.

Double reduction gearing permits of larger ratios between the revolutions of turbine and propellers being obtained without excessive size of gear wheel, so that even in very slow-speed vessels the turbines may be designed to run at the speed for maximum economy. The increased efficiency obtainable with a double reduction scheme and higher speed

turbines as compared with turbines and single reduction gears is in some cases as much as 7 per cent.

From first to last a considerable advance has been made in the efficiency of the steam turbine. In 1892 a turbine of 100-kilowatts gave a steam



consumption of about 27 pounds per kilowatt, whilst later in larger sizes, say of 3,000 kilowatts, a steam consumption of under 15 pounds was obtained. In 1907 there was a further reduction to 13.2 pounds per kilowatt, and at the present time steam consumptions for large land installa-

tions as low as 10.3 pounds per kilowatt-hour with 275 pounds steam pressure and 200 degrees of superheat, which is equivalent to about 7.5 pounds per brake horsepower, are being obtained.

In marine work, for the earlier turbine vessels the steam consumption of the turbines worked out at about 15 pounds to 16 pounds per shaft horsepower, and in later direct-driven turbines under 12 pounds per shaft horsepower was obtained, whereas at the present day, with double reduction gearing and reaction turbines, consumptions of under 10 pounds per shaft horsepower can be obtained with saturated steam, and under 8 pounds per shaft horsepower with a superheat of 200 degrees F. From these figures it will be readily seen that a remarkable advance has been made in increasing the efficiency of the turbine, the steam consumption being about one-half of what it was eighteen or twenty years ago.

By the courtesy of several shipowners Table A has been prepared, giving particulars of the performances of vessels with Parsons geared turbines, as compared with sister ships fitted with reciprocating engines.

There are several varieties of turbines now in use, some of them only, however, to a limited extent, but within the scope of the paper it is not possible to make more than a few general remarks.

The various turbines may be classed under the headings "Impulse" and "Reaction," and in some cases a combination of the two. The reaction class has low steam velocities. In modern practice the blades of a reaction turbine are of a reasonable length and the maximum possible efficiency is higher than obtainable with any other type.

The question of maintenance of efficiency and economy in wear and tear under long continued service is a very important one to shipowners. The reaction turbine is the only system which has been tried over a long period of years in mercantile vessels. The result of such experience points to the fact that, with ordinary care, the economy of the reaction turbine is fully maintained after years of running. This maintenance and economy is due to the low steam velocities employed. It has been found that with high steam velocities there is a tendency for the blades to suffer erosion, probably caused by drops of condensed water mingled with the steam, the action increasing rapidly with increase of velocity. Blades which have been eroded show a falling-off in efficiency, but this erosive action is reduced by the adoption of superheated steam.

As regards the life of gearing, there appears as the result of experience to be no reason why it should not have a very long life, provided, of course, that ordinary precautions are taken in the manufacture and cutting of the gears, and that efficient lubrication is provided. In this connection it may be mentioned that the gearing of the *Vespasian* shows practically no signs of wear. The *Cairnross*, which was the next vessel to the *Vespasian* to be fitted with geared turbines, has unfortunately been sunk by enemy action. The *Mahanada* has now been four and three-quarter years on continual service, and from a report made quite recently the gearing is "in as good a condition as when it first left the builders' works"; from inquiries made from the owners of this vessel, the turbines and gearing during this period have not cost anything for upkeep.

On land, geared turbines have been adopted for certain work in connection with the generating of electricity. In direct-current working before the introduction of gearing, difficulties were met with owing to the fact that dynamo speeds were too low for direct coupling to turbines of maximum efficiency. Owing to the limitation of dynamo design, the turbine speed had in some cases to be reduced to less than half the speed corresponding to maximum efficiency, and in such cases mechanical gearing has been employed with considerable advantage, an economy of 15

per cent having been effected. In the case of alternators of fair size the speeds are usually such that direct coupled turbines of maximum efficiency can be used.

One of the earliest applications of gearing was that of driving a continuous running plate rolling mill for the Calderbank Steel Works of Messrs. J. Dunlop & Co. This gearing, which was of the double reduction type, and driven by a turbine of 750 horsepower, has remained in continuous operation to the present time, and a recent examination showed no signs of wear in the teeth.

Gearing has opened up a new field for the steam turbine in the driving of cotton, jute, paper and other mills.

A number of mills, both in Great Britain and in the Colonies have adopted this method of driving with most satisfactory results, both from the point of view of economy and reliability.

In conclusion, it may be safely said that the introduction of the steam turbine, has added to the progress of the world, since the turbine has considerably cheapened mechanical power on land and sea, and has rendered possible speeds greatly in excess of those obtainable with reciprocating engines. Already it has entirely superseded the reciprocating engine for war vessels and for high speed and modern mercantile vessels, and bids fair in the near future to being applied on an extensive scale for low-speed vessels, in view of the substantial economy in coal consumption obtainable thereby.—"Engineering."

SCIENCE AND ITS APPLICATION TO MARINE PROBLEMS.*

By Professor J. C. McLennan, O.B.E., F.R.S., Scientific Adviser to the Admiralty.

Introduction.—In the great struggle which has just been brought to its close the introduction and use of the submarine as an agent of unrestricted warfare by our former enemies, constituted a menace of paramount and vital importance to the safety and welfare of the British Commonwealth and of its allied nations.

As early as 1915, a Board of Inventions and Research was set up in the Admiralty under the presidency of Admiral of the Fleet the Lord Fisher of Kilverstone, G.C.B., O.M., G.C.V.O., LL. D., for the purpose of developing anti-submarine devices and systems capable of coping with the menace. This board consisted of the President and Professor Sir Joseph J. Thomson, O.M., Pres. R.S.; the Hon. Sir Charles Parsons, K.C.B., F.R.S.; Sir George T. Beilby, F.R.S.; and Vice-Admiral Sir Richard Peirse, K.C.B., K.B.E., M.V.O.; and it was assisted by a panel of the ablest scientists available, as well as by a secretariat and series of subcommittees of leading naval officers and scientific experts.

In the autumn of 1917, after exhaustive researches had been carried out by this board, it was finally realized by all that the submarine problem was one of the most difficult ever presented to science for solution. It became clear that it was necessary to introduce into service practically a new system of physical science and engineering, and the Admiralty, at the instance of the Right Hon. Sir Eric Geddes, G.B.E., K.C.B., M.P., First Lord, decided to broaden the field from which scientific and engineering talent could be drawn, and set up within the Admiralty a Department of Research and Experiment under the direction of Mr. Charles H. Merz. The Board of Invention and Research was incorporated in this department,

*Paper read before the North-East Coast Institution of Engineers and Shipbuilders, at Newcastle-upon-Tyne, July 10, 1919.

and Anti-Submarine Committees were constituted in such important industrial and technical centers as the Lancashire and Clyde districts.

In the naval service, too, a special Anti-Submarine Division was organized under Captain W. W. Fisher, C.B., R.N., and in the Department of Torpedoes and Mines under Rear-Admiral the Hon. E. S. Fitzherbert, C.B., and later under Captain (nor Rear-Admiral) F. L. Field, C.B., R.N., provision was made for greatly extending by research, experiment and trial the capabilities and use of improvements in mines and torpedoes as aids to naval warfare. Steps were also taken to promote the closest and most efficient co-operation between the departments of the naval service and those of the civilian scientific staff.

Extensions were also made in their organizations by our Allies, and as a result of this great effort it may be stated that by the late summer of 1918 it became clear to those associated with the movement, that the submarine problem was well in hand from a scientific point of view, that the character of the means of coping with the menace in an effective manner was clearly defined, and that the elimination of the pest by the service was only a matter of a few months' time.

At the time the Armistice was signed, only a few of the most powerful and effective devices developed were actually in service, but the results which were obtained by means of these for the short time they were in use were proportionately great. It is but right to say that the destruction of by far the great majority of the enemy submarines which were rendered innocuous was accomplished by the naval service by the use on an enormous scale of the ordinary methods available to it before the war began, or which were rendered available to the service in the early years of the war. Potentially, however, science by its advances has greatly extended the basis of service methods, and though much has yet to be done in the field of development of particular systems and devices it can be said, speaking conservatively, that the submarine in its present stage of development and in the method of its use hitherto adopted by our former enemies has been overcome as a menace to our national safety and well being.

For several reasons one in my position cannot give details of a number of the measures taken and means of destruction adopted in the anti-submarine campaign, but it is in the national interest to foster and stimulate co-operation by our people in the development of science and of its application to naval and other marine problems. In the following paper, therefore, a brief survey will be given of some of the most notable advances made in dealing with certain naval and marine problems presented to us on the naval side during the war.

ANTI-SUBMARINE MEASURES AND DEVICES.

(*a*) *Listening Devices.*—As the power of the submarine in its attack is due to its invisibility, it is clear that in countering its methods must be employed which will reveal its presence and give definite information regarding its movements. Of all the physical disturbances emitted or produced by a moving submarine the pressure waves set up in water by vibrations having their source in the vessel are the ones which are propagated to, and are detectable at, the greatest distance. Efforts were therefore directed from the first to the development of listening devices. The investigations in this field were exceedingly ramified and were pressed with great vigor.

As a result great improvements were made in hydrophones. In the development of these devices microphones and magnetophones of ex-

ceedingly high sensitivity were realized, and after enormous labor, ways and means were devised for standardizing their construction and their functioning. Hydrophones were constructed and put into service which were suitable for use in water of moderate depth, and other types were made which could be used in water of great depth. In one particular type of instrument modifications and attachments were introduced which enabled one to detect with it the direction of bearing of a source of sound with a fair degree of accuracy.

But probably the method of determining the direction of a source of sound waves in water which has proved to be the best is founded on the fact that the sound wave is in the same phase at all points of its wave-front. Thus, if there were two hydrophones, in themselves non-directional, placed in the path of the incoming sound, they can be used for finding the direction of the origin of the sound if the phase difference between the sounds received can be detected. There are two ways of doing this: the "binaural" method and the "sum and difference" method. The binaural method depends on the fact that if the sound from one receiver is conveyed to one ear and that from the second to the other ear, the impression is formed that the sound comes from a certain direction, and this direction, as interpreted by the sensations experienced, changes as the phase difference is altered. It can be brought to a certain position with respect to the listener (say, to the position directly in front) either by rotating the two receivers about an axis, or by introducing an artificial delay in some form of "compensator." This binaural method, which has great interest from a psychological and physiological point of view, has been the subject of much work on the part of both British and American scientists, and in the antisubmarine campaign it was found to be of very considerable service.

In the "sum and difference" method the impulses from the two receivers are united before reaching the ear, the combined effect observed being a maximum when there is no phase difference between the waves and a minimum when the phases are in opposition.

The French Navy has developed a hydrophone or listening device known as the Walser gear, which it has found very efficient. In this device two sets of a number of sound-receiving plates or studs are mounted on two areas which are convex to the sea, one on each side of the keel, on the underside of the hunting ship's hull. By this gear the sound impulses which they receive are not transmitted by tubes, but are given freely to the air inside the ship. These sound impulses are concentrated in a focus whose position in space depends on the direction from which the sound comes. The position of the sound focus is determined by means of a funnel or trumpet, from the contracted end of which a flexible tube leads to the listener's ear. The great advantage possessed by the Walser gear is that it enables one to distinguish by differences in the positions of the sound foci between sounds coming from a distant source and those generated in the water which have no specific directional feature characterizing them. The gear can therefore be used by chasing ships when moving with fair speeds.

With most hydrophones and many listening devices, ship's noises and water noises generally seriously interfered with their effective use, and in practice the chasing ship is compelled to stop at intervals and listen when not under way. This means that in many cases the quarry is lost.

This defect was overcome in a measure by towing a directional hydrophone encased in an artificial "fish" behind a chasing ship. By adopting stream-line formation for the towed body and suitably supporting the hydrophone, water noises are fairly well eliminated. Moreover, as the

"fish" can be towed at a considerable distance behind the chasing vessel, many of the sounds emitted by the latter do not reach it, and others which do arrive are received with weakened intensity.

It is needless to say that the development of sensitive listening devices received a great impetus by the use of thermionic amplifying valves.

Although an enormous expenditure has been made in time and effort to perfect the hydrophone and other listening devices, it is realized that such instruments possess an inherent defect. If the submarine can be made noiseless in motion, this method of countering it becomes ineffective. Even now the range of hearing is not more than 100 yards in the case of modern submarines moving at 2 knots or 3 knots.

(*b*) *Echo Methods.*—Owing to the fact that it was found possible under certain conditions to render the propulsion of submarines practically silent, it became necessary to look in other directions for fundamental methods of detecting them. A system of detection which is full of promise involves the use of a beam of sound waves sent out by a chasing ship in a manner analogous to the use of a searchlight. With such beams of sound waves it is possible to sweep the seas, and when an object of sound such as a submarine happens to come within the beam, the sound waves are reflected and echo effects are obtainable. The character of the beam is, of course, determined in large measure by the frequency of the waves constituting it. The method has been employed with great success, and promises to be a very helpful agent. It can be used by chasing ships traveling at all speeds, and when applied with certain restrictions and definite characteristics it enables one to pick up and close on a submarine situated more than a mile away. The method, it is obvious, is applicable to the locating of minefields and other obstacles to navigation as well as to submarine chasing.

(*c*) *Magnetic and Electro-Magnetic Detection.*—Magnetic detectors usually require the movable system to be poised or pivoted. They can therefore be used as yet with only a moderate degree of satisfaction in towed bodies or in vessels subjected to violent mechanical disturbances. The range at which magnetic effects can be detected is moreover comparatively short. As a result of these defects the use of magnetic detection is somewhat circumscribed. Such instruments can, however, be used under certain conditions, and in particular sea areas with great effect. In the war very considerable results were actually obtained by their use.

The range at which electro-magnetic detection can be applied is greater than is possible with magnetic detection, but the method is, however, essentially a short-range one, and in many of the forms in which it has been worked out it cannot be used with success at distances greater than about 300 yards or in depths greater than about 100 fathoms.

(*d*) *Leader Gear.*—An important application of an electro-magnetic effect which was developed during the war is found in what is known as Leader gear. This gear consists of a cable laid on the bottom of the sea along the course of a narrow tortuous channel leading into a harbor or through a minefield. If an alternating electric current be passed through such a cable it is possible by means of delicate devices installed on a ship to obtain either aural or visual indications of the presence of such a cable, and by these indications the ship can be guided in safety in fog or darkness at speeds as high as 20 knots almost with as much precision as a tram-car by a trolley wire over a railway. Experiment has shown that it is a simple matter to apply this method in water of suitable depth for distances as great as 50 miles or longer.

(*e*) *Invisible Signalling.*—Research has shown that it is possible under certain conditions to utilize polarized light or ultraviolet and infra-red radi-

ations for secret signalling. With the last-mentioned type of radiation especially valuable results are obtainable over considerable distances, even in the presence of light fogs. Where it is not advisable to use wireless communication between chasing ships, infra-red signalling is of special value.

(*f*) *Wireless Telegraphy and Telephony.*—One of the most remarkable developments which have taken place in the war is in the field of wireless telegraphy and telephony. By the use of oscillating thermionic valves especially great progress has been made. It is now possible to hold conversation with ease between a land station or a ship and an airship or seaplane over considerable distances, and by this means observers on aeroplanes or aircraft can also converse with one another. With high-power installations it has been demonstrated that wireless telephonic communication can be maintained on the sea over hundreds of miles.

On the directional side of "wireless" great advances have also been made. If an aeroplane, an airship, or surface ship should send out continuously for a short interval a series of ether waves, these waves can be picked up over long distances by devices installed in a land station, the direction of the source of these ether waves can be ascertained, and in a minute or two the land station can give the observer of the emitting source his bearing within two degrees relative to the land station. With two land stations it is possible to obtain cross bearings, and the latitude and longitude of the sending air or surface ship can be determined with a high degree of accuracy.

With directional devices installed on ships it will be possible for two ships whose positions are known to communicate its true position to a ship enveloped in a fog and situated several hundreds of miles away. It is obvious, therefore, from the few illustrations which have been given that directional wireless will find a wide field of usefulness in the future in connection with the subject of navigation.

(*g*) *Explosion Pressures.*—In the early days of the anti-submarine campaign, a method of destroying submarines whose approximate location was known, was by the employment of depth charges. To use this means it was necessary first of all to know the neighborhood in which the submarine was located, and then the chasing ship would rush to the spot and drop or throw to some distance charges of explosive which detonated when they reached a definite distance below the surface of the water. The necessity of knowing the destructive zone of any given type of depth charge soon became evident, *i.e.*, to determine the radius from the exploding charge, within which a submarine would be successfully destroyed. The same information was important in the laying out and use of mine-fields. Investigations were undertaken to determine what pressures were generated by charges of different sizes and types at various distances from the place of detonation. The nature of the pressure wave was particularly important, for upon it depends the "killing power" of the charge. The laws which govern the alteration in form and power of waves generated by these explosions had to be determined in order to employ depth charges and mines in the most effectual and economical manner. The accurate determination of the velocity of propagation of the explosive waves generated was also of importance in distributing the charges, for if waves from two different sources arrive at the object in different phases, the effective crushing power may be considerably altered. When we know what the effect of different charges at various distances is—what type of pressure wave, whether a sudden intense blow lasting a few ten-thousandths of a second, or a sustained pressure during some thousandths of a second, or a series of less intense shocks, has the greater effect in de-

stroying the submarine when under water—then we shall know how best to lay out our minefields, and what size and type of charges are the best and most economical to utilize under the various situations which may arise.

A most elaborate investigation of the characteristics of explosion pressure waves has been carried out for the Admiralty by Mr. H. W. Hilliar. By this method the pressure is allowed to act on one end of a steel piston, and measurements are made of the velocity of the piston as it passes a series of points at known distances from its starting point. From these measurements it is only a mathematical operation to extract the acceleration of the piston at different times from the moment when it began to move; in other words, the time history of the pressure acting on the piston. It is not actually feasible to measure the velocity of a single piston at different stages of its travel, but you get the same information if you let the pressure act on a series of pistons and measure their velocity after they have travelled different distances. The velocity is found from the extent of crushing of a copper plug which the piston strikes when it comes to the end of its travel.

As an illustration of results obtained by the method it may be stated that with the measuring gauges at a distance of 50 feet from a 300-pound detonating charge of amatol placed about 50 feet below the surface, the corresponding time-pressure curve showed that the maximum pressure, 0.80 ton per square inch, was reached almost instantaneously; that the pressure fell to one-quarter of its maximum value in 1/1,000 of a second, and practically faded away after 5/1,000 of a second. In the course of the investigation it was shown that the pressure waves are reflected from the water surface as waves of tension. The effect at any given point in the neighborhood of the explosion is therefore due to the superposition of a direct pressure wave from the charge and a reflected tension wave from the surface: both travel with the velocity of sound (4,900 feet per second) and the tension wave follows the pressure wave after an interval determined simply by the difference in the length of the direct and reflected paths from the charge to the point in question.

It has been found that the pressure from a large charge is more intense and more sustained than that from a small one, the two being connected by the following rule. If one charge has twice the linear dimensions of another (eight times the weight) the maximum pressure at a given distance from the larger charge will be the same as at half the distance from the small charge and will be twice as sustained, i.e., will take twice as long in falling to any given fraction of the maximum.

When explosions of gunpowder, for example, were investigated, it was found that the pressure rose much more gradually than when charges of amatol or T.N.T. were used. With this explosive very low maximum pressures were obtained and the corresponding pressure waves were considerably prolonged.

Another method of investigating such pressures, which was suggested by Sir J. J. Thomson and applied by Mr. D. A. Keys, consists in the employment of the phenomenon long known to scientists, namely, that certain crystals become charged with electricity when subjected to pressure. The amount of the charge produced is proportional to the pressure applied to the crystals, so by having a suitable arrangement for measuring this charge and its variation with time, a complete record of the variation of pressure with time is obtained by placing the crystal detector at any given distance from the exploding charge. Since the duration of the wave in passing over the crystal or engulfing a submarine is only a few thousands of a second and the pressure generated may be of the order of half a ton

or more per square inch, one can readily imagine the difficult nature of the problem in hand. But by making use of the inertia of a beam of cathode ray particles and employing the fact that they carry negative charges and are deflected by electrostatic and magnetic fields, it has been possible to obtain records of the variation of such pressures with the time. The electrons affect a photographic plate, *i.e.*, they leave an impression on the plate where they strike it. This additional fact has made it possible to determine the change in pressure of the wave from the instant the charge is fired and at as small intervals as we please afterwards. Changes which take place in 1/100,000 of a second have been recorded by this means. Already the method has revealed a number of facts about the nature of the pressure wave produced by exploding charges and the importance of such results in the laying out of minefields and the employment of depth charges can hardly be over-estimated.

(*h*) *Sound Ranging.*—In the course of our investigations of the characteristics of pressure waves generated by the explosion of charges in the sea, it was found that when a hydrophone was used to pick up the waves a good record could be obtained by the explosion of a number 9 detonator at least 2 miles away. The explosions due to charges of 2 pounds of T.N.T. have been recorded at 14 miles, and might have been recorded at far greater distances judging from the strength of the signals received. The explosions of 300-pound depth charges have been recorded up to 200 miles, and it is probable that with charges of moderate amount explosions occurring as far away as 500 miles can be readily recorded. Based on these results a system of sound ranging under water was developed. Four hydrophones were laid out 5 miles apart along a base line in deep water a mile or two from the shore, and in addition two pilot hydrophones were placed along a line at right angle to the base line, the one 5 miles out and the other at twice that distance. Cables were laid from the hydrophones to a recording instrument situated in a shore station. Four of these stations were installed at different places along the east coast of the British Isles, and other stations are now in progress of installation. With such sound-ranging systems the shock of distant explosions occurring under water affect the various hydrophones in turn, and as time intervals can be read to two or three thousandths of a second with the apparatus now in use, it is possible to measure with accuracy the time intervals between the times of arrival of a sound wave at the different hydrophones. With the measurements of these time intervals it is a simple matter to deduce the position of the point at which the explosion setting up the wave is located. Up to 50 miles the location of an explosion under water can be determined to within a few hundred yards by a single station, but for accuracy the co-operation of two stations would be necessary to locate explosions at greater distances. Within operable ranges a ship can be given its position by sound ranging more accurately than by directional wireless or by any other known method. Explosions of mines or torpedoes at any point in the North Sea can easily be located by stations situated in Great Britain. In the war, during the bombardment of the Belgian coast it was a common thing for a monitor to proceed in a fog to a position some miles from the coast and by dropping depth charges have its position accurately determined from stations on the coast of England. So accurately was this done that it was found when the monitor's guns were trained in selected directions objectives several miles inland could be hit with regularity and with a minimum expenditure of ammunition.

(*i*) *Helium.*—Shortly after the commencement of the war it became evident that if helium were available in sufficient quantities to replace

hydrogen in naval and military airships, the loss in life and equipment arising from the use of hydrogen would be enormously lessened. Helium is most suitable as a filling for airship envelopes in that it is non-inflammable and non-explosive, and, if desired, the engines may be placed within the envelope. By its use it is possible to secure additional buoyancy by heating the gas (electrically or otherwise), and this fact might possibly lead to considerable modifications in the technique of airship maneuvers and navigation. The loss of gas from diffusion through its envelope is less with helium than with hydrogen, but on the other hand the lifting power of helium is about 10 per cent less than that of hydrogen.

Proposals had been frequently put forward by scientists in the British Empire and in enemy countries regarding the development of supplies of helium for airship purposes, but the first attempt to give practical effect to these proposals was initiated by Sir Richard Threlfall, who received strong support from the Admiralty through the Board of Invention and Research.

It was known that supplies of natural gas containing helium, in varying amounts, existed in America, and it became evident from the preliminary investigations made by Sir Richard Threlfall and from calculations submitted by him as to the cost of production, transportation, etc., that there was substantial ground for believing that helium could be obained in large quantities at a cost which would not be prohibitive.

The writer was invited by the Board of Invention and Research in 1915 to determine the helium content of the supplies of natural gas within the Empire, to carry out a series of experiments on a semi-commercial scale with the helium supplies available, and also to work out all technical details in connection with the large-scale production of helium and the large-scale purification of such supplies as might be delivered, and become contaminated with air in service.

In the course of these investigations which were carried out, with the co-operation of L'Air Liquide Company, it was found that large supplies of helium were available in Canada, which could be produced at a cost of about 1*s.* per cubic foot.

In the spring of 1917, when the United States of America had decided to enter the war on the side of the Allies, and after the investigations referred to above were well under way, proposals were made to the Navy and Air Board and to the National Research Council of the United States of America to co-operate by developing the supplies of helium available in the United States. The authorities cited agreed to co-operate with vigor in supporting these proposals, and large orders were at once placed by them with the Air Reduction Company and the Linde Company for plant, equipment, cylinders, etc. The Bureau of Mines also co-operated by taking steps to develop a new type of rectifying and purifying machine. By July, 1918, the production of helium in moderate quantities was accomplished, and from that time forward the possibility of securing large supplies of helium was assured.

Concurrently all practical details of the construction of helium-borne airships and of the navigation of this type of craft were developed by the Airship Production Section of the Navy. At the same time, under the direction of the writer, plans were prepared and steps were taken to erect and equip a station for purifying the helium which might become contaminated in service.

Experimental investigations were also initiated with the object of developing all possible technical and scientific uses of helium. In particular, balance, spectroscopic and electrical methods of testing the purity of the gas were worked out, studies on the permeability of balloon fabrics

to hydrogen and helium were commenced, and experiments were begun to exploit the use of helium in gas-filled incandescent lamps; gas-filled arc lamps, and in thermionic amplifying valves. The equipment provided for the purification of contaminated helium in large quantities supplied the major portion of the apparatus required to liquefy helium, and arrangements were therefore made to produce this gas in a liquid form for the purpose of carrying out low-temperature researches.

The advances actually made at the time the armistice was signed warrants the opinion that by the present time, had the work projected been completed, supplies of helium at the rate of 2,000,000 cubic feet a month would have been produced within the Empire and the United States at a low cost, helium-filled aircraft would have been in service, and great progress would have been made in exploiting the technical and scientific uses of this gas.

Before the war a proposal to utilize helium as a filling for airships would have been viewed, even by most scientists, as impossible, but thanks to the enterprise, enthusiasm and initiative of the Navy, backed by imagination, a suggestion—at one time considered to be chimerical—has today become a realization.

DEFENSIVE MEASURES.

From time to time publicity has been given to the steps taken by the British Navy in co-operation with the Navy of the United States of America to close to the passage of submarines such sea areas as the northern portion of the North Sea and the Straits of Dover.

At the time of the signing of the armistice, this stupendous task was well advanced. The material used consisted largely of ordinary contact mines, which were used in vast numbers and at the expenditure of enormous labor and capital.

It can be stated now, however, that concurrently with the installation of this system of defense, other systems of dealing with these and similar areas were worked out, which involved the use of more subtle mechanisms in quantities which were vastly smaller in amount.

Today it is scientifically possible and practicable with a comparatively small amount of material to close effectively to the passage of submarines by either automatic mechanisms or by controlled ones such sea areas as the Bristol Channel, the water stretch between the Mull of Cantyre and the north coast of Ireland, the Straits of Dover, the sea area between Belgium and Denmark, the Cattegat and Skager Rack, and the greater portion of the North Sea between the Orkneys and Norway.

This will serve to show, in a measure, the part science has been able to play during the war. Had the knowledge we now possess been available at the opening of the war, we should have been spared much inconvenience, suffering and anxiety, for the submarine menace, at one time threatening and uncomfortably dangerous, would never have been able to materialize.

APPLICATIONS OF SCIENCE UNDER PEACE CONDITIONS.

Under peace conditions many important technical systems and devices brought forward during the war will find immediate application as aids to navigation.

By means of directional wireless systems, ships or aircraft in the English Channel, the North Sea, or in the eastern and western portions of the North Atlantic, can be given their positions when prevented from getting it by the existence of fogs or unfavorable weather. By means of sound ranging it is possible to fix the positions of light vessels, buoys which indicate channels and obstructions, such as sunken ships. Ships steaming in

fog up the Channel or approaching the shores of Nova Scotia, Newfoundland or Labrador, can be given their positions with accuracy for ranges up to as much as 500 miles. Seaplanes and aircraft in distress in the neighborhood of the British Isles or near the coasts of America can call for help and be located when wireless gear becomes inoperable by simply dropping depth charges.

In hydrographic work generally sound ranging will be of the greatest service for surveys can be made and investigations of seabeds carried out in fogs as well as in fair weather without the delays which have been experienced in the past. The positions of the localities being investigated can always be determined in a few minutes when once a sound-ranging station has been established on some shore within reach.

By Leader gear laid in such areas as the River St. Lawrence, the entrance to the Thames or to Halifax Harbor, the Straits of Dover, etc., in and out lanes of traffic can be organized which can be maintained with ease in fogs. The Echo methods to which reference has been made can be used for founding, for locating icebergs, surface vessels and rock-bound coasts in a fog as well as for locating submarines.

Helium which was originally produced as a filling for airships can be utilized for the production of illuminating agents and for providing a means of investigating the fundamental properties of matter, at the lowest temperatures attainable by man. Developments in internal-combustion engines, in electric drive, and in the fuel values of new materials which were to be corporated in the Navy, will be of enormous value to the mercantile marine in the future.

Advances made in wireless telegraphy and telephony and in secret signalling by specific types of radiation for war purposes, will also prove of great service under peace conditions by providing us with novel, efficient, and less costly methods of communication.

PROPOSED SCIENTIFIC ESTABLISHMENTS FOR THE FUTURE.

With a view to developing and extending the scientific results which were obtained under stress of war, the Admiralty has recently put forward proposals for the permanent establishment of a Department of Research and Experiment within the Navy.

Plans have been formulated for the erection of a Centrals Research Institution for the investigation of first principles and for carrying on researches of a fundamental and pioneer character. Steps have already been taken to organize a sea experimental station and to provide buildings and equipment for an engineering laboratory, a wireless and signal school, and a torpedo and mining school in place of the "Vernon." It is believed that these institutions will prove of great value in developing not only means of increasing the efficiency of the Navy, but in providing aids to navigation for our mercantile marine.

The initial expenditure for buildings and equipment will be large, but it seems evident that an ample financial return will in a short time be obtained for the nation from profits accruing from a lowering of the rates of insurance and from a reduction in the cost of transportation. If we could by the use of such aids to navigation as have been referred to above prevent two or three wrecks per year, or lower the time of passage between Great Britain and Canada on the average by one day per voyage per ship through the fog-covered areas in the neighborhood of Newfoundland, sufficient funds would be saved in a year or two to cover the whole cost of the expenditure on scientific and experimental establishments and on the prosecution of the researches and investigations foreshadowed. —"Engineering."

OIL-FUEL INSTALLATION IN PASSENGER STEAMSHIPS.

New Board of Trade Instructions.

In view of the increasing use of oil fuel for boilers, the Board of Trade has issued for the information and guidance of surveyors the following instructions relating to oil-fuel installations on passenger steamers. Excepting the minimum flash point of the oil, which has been reduced from 185 degrees F. to 175 degrees F., the instructions represent, in a general form, the practice which has been followed by the Department in dealing with the numerous individual cases which have been submitted to them.

Flash Point.—The flash point of the oil-fuel should be 175 degrees F., or upwards, as determined by Abel's close test. With each supply of oil taken on board, a written guarantee that the flash point is not below 175 degrees F., signed by a responsible official of the firm supplying the oil, should be furnished. As opportunity offers, the chief engineer should make check tests of the flash point of each supply taken on board, in order that he may have accurate knowledge of the class of oil he is using.

Storage Tanks, Etc.—Oil fuel of the above description may be carried in cellular double-bottoms under engine and boiler compartments, and under ordinary holds; also in peak tanks, deep tanks, and bunkers of approved construction, particulars of which should, in the first place, be submitted for consideration. Provision should be made for the expansion of the fuel in tanks. If the storage and settling tanks are to be constructed to the requirements of either Lloyd's Register, the British Corporation, or Bureau Veritas Registry, a copy of the detailed plans approved by the classification society should be forwarded to the Principal Ship Surveyor for consideration. In other cases, fully detailed plans of the proposed construction, including riveting, should be submitted for approval before the work is taken in hand. Double-bottom compartments used for oil-fuel storage are to be fitted with center divisions. In other storage tanks, suitable wash plates are to be fitted. Where oil fuel is carried in wing spaces on each side of the ship, these spaces must be connected by means of suitable pipes or ducts fitted as low down in the ship as practicable, in order to comply with the requirements of paragraph 61 of the "Instructions Relating to the Construction of Passenger Steamships."

Suitable air and filling pipes are to be provided. The open ends of the air pipes should be led to a situation above deck where no danger will be incurred from the discharge of oil vapor therefrom when the tanks and bunkers are being filled, and each should be furnished with a wire-gauze diaphragm which can readily be removed for cleansing or renewal.

Every oil fuel tank is to be fitted with a sounding pipe led to a suitable position above the crown of the tank. Where air or sounding pipes pass through the cargo holds, arrangements must be made for effectually protecting them from being damaged. No short sounding pipes should be passed either in tunnels or elsewhere.

Provision should be made to prevent over-filling the tanks or bunkers, and gutter ways and coffer dams should be fitted where required. If fresh water is stored in a compartment adjacent to an oil tank, a coffer dam is to be fitted or other effective means taken to prevent the water from being contaminated. If double bottoms under cargo holds are used for the storage of oil fuel, efficient means must be provided by wells or gutter ways to prevent leakage from any oil fuel compartment coming in contact with the cargo, and to ensure that such leakage shall have free drainage into the limbers or wells.

In steamships trading in climates where the cold may cause the oil to become viscous, heating coils should be fitted near the open ends of the

suction pipes in the storage tanks, to render the oil sufficiently fluid to flow freely through the pipes.

Tests.—Every bunker or storage tank is to be tested by filling with water to a head at least 1 foot above the highest point to which oil has access in pipes, hatchways, or elsewhere. In new steamships, the double bottom is, however, to be tested with a head of water up to the bulkhead deck, as required by paragraph 78 of the "Instructions Relating to the Construction of Passenger Steamships."

Settling Tanks.—If the oil is heated in the settling tanks, in order to facilitate the separation of the oil from the water, open drains should not be allowed unless satisfactory precautions are taken that the oil shall not be heated to a temperature at which oil vapor may be given off. Preferably, the drainage should discharge directly overboard, by gravity, or it should be pumped overboard.

Insulation, Etc.—The boilers should be suitably lagged. The clearance space between the boilers and the sides of the storage tanks or bunkers in which oil fuel is carried should be adequate for the free circulation of air necessary to keep the temperature of the stored oil well below the flash point; and bunkers which overhang the boilers in close proximity thereto should be fitted with shield plates.

Funnel Dampers.—Funnel dampers should not, as a rule, be fitted, but, if fitted, they must be provided with a suitable device whereby they may be securely locked in the fully open position. On ships fitted out for the use of oil only, dampers should not be fitted.

The smoke-box doors should be shielded, well fitting, and kept shut.

Pumps.—The pumps for the oil fuel system should be entirely separate from the feed, bilge, and ballast pumps and connections, and should be provided with efficient escape valves which must be in close circuit, *i.e.*, discharging to the suction side of the pumps.

Pipes.—The oil pressure pipes should be solid drawn, and those for conveying heated oil should be placed in sight above the platform in well-lighted parts of the stokehold or engine room. The coupling flanges of the pressure and other oil pipes should be machined, and thin material should be used for the joints so that the flanges may be, practically, metal to metal.

The pipes should be tested to at least double the working pressure.

Cocks and Valves.—All oil fuel suction pipes from storage or service tanks placed above the double-bottom should be furnished with cocks or valves secured to the tanks and so geared that they may be shut off from the deck above or from a compartment other than the one in which they are situated, as well as from the latter compartment. If the discharge pipes are not connected to the tanks near the top, they should likewise be provided with cocks or valves similarly operated, or else non-return valves should be fitted.

Cocks or valves should also be interposed between the pumps and the suction pipes, in order that the pipes may be shut off when the pumps are opened out for overhauling.

It is desirable that no gauge glasses be fitted either to the storage tanks or to the service tanks; but, if suitably protected, gauges of approved special design, or others having flat glasses of substantial thickness and cocks whereby they may be shut off in the same manner as required above for the suction cocks on these tanks may be allowed.

Heaters.—If steam is used for heating the oil fuel, the exhaust drain should discharge the water of condensation into a tank where it can be seen whether or not it is free from oil.

Lighting.—No exposed lights should be placed in any space where oil

vapor might accumulate, such as the pump room or stokehold, which should be illuminated by means of an electrical installation, preferably on the two-wire, two-conductor system, when the electrical pressure exceeds 110 volts, and no switches or fuses should be placed in any such space. The electric lamps should be protected by wire guards if considered necessary.

Electric torches should be provided for examining the bilges, pipes, etc., for leakage, and self-contained, battery-fed lamps, similar to those used in fiery mines, may be used in dangerous places. Portable lamps supplied with current through flexible cables should not be permitted.

Precautions Against Fire.—In addition to the usual water service and hose and pipe conductors, one of which might be provided with a nozzle having perforations in it to spray the discharge, pipes perforated for the emission of steam into the gutter-ways or lower parts of the boiler room should be supplied, the control of steam supply being outside the boiler room. The boiler room should contain a receptacle for holding about 5 cwt. of sand, and suitable scoops for distributing the sand should be provided, together with chemical fluid extincteurs for use in the boiler room, pump room, or engine room.

Contiguous Passenger or Crew Spaces.—Neither passengers nor crew should be berthed or accommodated in a space adjoining an oil bunker and separated therefrom only by the containing bulkhead of the bunker. No objection need be raised, however, to passenger or crew spaces situated on a deck forming the crown of an oil fuel space, provided that the deck is constructed thoroughly oil tight, that there are no manholes or hatchways to the oil spaces situated within the quarters, that the flooring is laid with a non-inflammable and non-magnesite composition at least 1½ inches thick, and that the spaces are especially well ventilated.

Outfit.—Thermometers and an apparatus for determining the flash point of the oil fuel should be supplied; also extinguishers, in the form of tubes with one end closed, for the torches used in igniting the oil spray at the burners.

Printed or typed instructions regarding the working of the system should be furnished for the guidance of the engineers; also a plan, suitably mounted, of the oil-piping arrangements.

Miscellaneous.—The escape of oil fuel heated to or above the flash point is most dangerous, and may result in an explosion or a fire should a naked light come into contact with the highly inflammable gas which is evolved.

Ample ventilation should be provided in engine, boiler and pump rooms, and also in all compartments adjacent to the storage tanks.

After lighting the burners, the torches should on no account be thrown away before carefully extinguishing by means of the appliances provided for the purpose.

Savealls should be provided under the pumps, heaters, and strainers where necessary, to catch oil which may be spilled when any cover or door is removed; and, likewise, at the furnace mouths when not protected by fronts which would intercept oil having escaped from the burners or becoming extinguished.

There should be no woodwork in the stokehold or compartment containing the settling tanks, and no wood or other combustible matter should be allowed to accumulate therein or in the vicinity of the fuel tanks.

Cleanliness Essential to Safety.—No oil should be allowed to accumulate in the bilges or gutter ways or on the tank tops. These parts should be washed out with a hose, having a conductor, at least twice a day, or oftener if required, and the wells should then be pumped dry. In order

that the gutter ways may be readily accessible for inspection and cleaning, it is desirable that the hold platform should be kept clear of all bulkhead plating.

Before any tank or bunker which has contained oil fuel is entered for inspection or repair, the oil should be entirely removed, and care should be taken that all oil vapor is also removed by steaming or by efficiently ventilating. Satisfactory tests of the atmosphere in the tanks or bunkers should be made to ensure safety against explosion before inspection or work in them is begun.

General.—If any difficulty arises in securing the adoption of the arrangements indicated in these instructions, the Surveyor should refer the matter for consideration by the Engineer-Surveyor-in-Chief.—"Steamship."

THE MEASUREMENT OF CONDENSER PERFORMANCE.

The extent to which the steam turbine is used at the present time for the propulsion of ships makes the problem of efficient condenser design one of the greatest importance. It is generally recognized by marine engineers that the best results from a steam turbine can only be obtained when the steam is allowed to expand to the lowest possible pressure before passing through the exhaust to the condenser, and hence, on turbine ships, it is customary to instal condensers which are capable of producing a higher vacuum than is the case on ships fitted with reciprocating engines. The production of an extra half inch in the condenser vacuum will result in a large increase in the power developed by the turbine, and it has become the practice to run the condensing plant on turbine ships so that the vacuum approaches to within about 1 inch of barometric height, or, on an average, to about 29 inches of mercury, while on vessels fitted with reciprocating engines, a 26-inch or 27-inch vacuum was generally considered as representing the limit above which it was not economical to proceed. It is not necessary at the present juncture to explain why it is that the turbine installation is always associated with a higher vacuum in the condenser, but rather to consider the various means which are adopted for producing this result, and to try to establish some basis of comparison in order to determine to what extent each method may be considered as an economical procedure.

Considering the condensing plant as comprising the condenser itself with merely its circulating pump and air pump, an increase in the vacuum can be obtained by increasing the size of the condenser or by fitting a larger air pump when dealing with a given quantity of exhaust steam. Merely increasing the size of the circulating pump tends to lower the temperature of the condensate with corresponding loss of efficiency of the boiler; although in dealing with a larger condenser a correspondingly larger circulating pump will of course be required. Hence, to produce a higher vacuum, we shall be faced with a larger expenditure of steam in both the air pump and the circulating pump, and the question arises as to how far these increases can be met before the saving in consumption in the main turbine is balanced by the increased consumption in the condenser auxiliaries. In addition to the factor of steam consumption, there is also the question of the increased weight and cost of the plant to be set against the saving in the cost of running it, and this is a point which must not be lost sight of when dealing with the increase in the size of auxiliary machines for any particular purpose.

Although no definite data are available as to when the simple condensing arrangement referred to above ceases to be economical, it is realized that in the production of high vacua some additional means must be adopted.

There are many systems of producing a high vacuum which have been tried in conjunction with turbine installations on board ship, and all of them achieve a certain measure of success. In one system we have what is termed an augmenter condenser by means of which the vacuum obtained in the main condenser is still further increased. In another system we have two air pumps, one connected to the bottom of the condenser and one to the top, and known respectively as the wet and dry air pumps. Yet, again, we have systems in which rotary pumps are employed in preference to reciprocating pumps, while there are many devices in which the exhausting effects of jets of either steam or water are employed in order to reduce the pressure in the main condenser. All these systems, as we have already said, succeed more or less in their object of producing a high vacuum, but it is very difficult to say at what expense in steam consumption this vacuum is obtained, and in any case the marine superintendent who is desirous of forming an opinion as to the relative merits of the different systems with which he is confronted, is at a loss to find a common basis from which the systems may be compared.

In measuring condenser performance there are so many variable quantities to be taken into account, that it is a difficult matter to find a common denominator in terms of which the performance of any plant may be expressed. Neglecting the question of the prime cost of the different systems, we might determine the number of pounds of steam condensed per pound of steam employed in the auxiliaries; but it must be remembered that these figures will only be of value when the pressure of the steam used in the auxiliaries is considered. Again, the temperature of the circulating water will materially affect the result, while the pressure, and hence the temperature at which the exhaust steam is condensed, must also be considered. But it is apparent that merely to say that a condensing system is capable of giving a vacuum of so many inches, or so much per cent of barometer, is not sufficient unless some indication is also given of the expense in steam consumption, and in other terms, with which this vacuum is obtained. The growing importance of the steam turbine for merchant ship propulsion and the consequent demand for condensing plant capable of yielding high vacua, renders it almost imperative that some means of expressing the efficiency of the performance of condensing apparatus should be devised. The above ideas have only been framed with a view to pointing out some of the difficulties of the question and to suggest a means whereby comparative figures might be obtained. But while the problem is one of great difficulty, it is one of great importance. Many systems of condensing are available. Many more will doubtless appear as the demand increases, and the superintending engineer will be called upon to make a choice as to which system is the most economical from all points of view. In helping him to make that choice, some basis of comparison is clearly desirable.—" Shipbuilding and Shipping Record."

WATERTIGHT DOORS IN MERCHANT SHIPS.

It is now possible to analyze experience during the great test of war on the question of watertight doors in ships. The controversy usually centers around the need or otherwise for doors in watertight bulkheads, and the difficulty has been expressed by saying that the question is one of the safety of the ship *versus* the convenience of the departments working the ship. It was abundantly demonstrated during the war that watertight doors when left open are a positive source of danger. The majority of vessels lost were sunk by the explosion of a mine or torpedo, and it was found that any bulkhead in the vicinity of the explosion was usually distorted to

such an extent that it was impossible to close the door. Actually it was found that if the door was open at the time of the explosion there was generally, with few exceptions, no time to shut it even if it had been in working order. It has often been proved that watertight doors are open when an unforeseen contingency arises. It has also been shown that it is not possible to rely upon a crew, however well disciplined, to close watertight doors in an emergency. Examples of this occurred during the war, where in many cases a watertight door was far enough from the center of the explosion not to be distorted, but only in isolated instances were such doors shut after the explosion occurred. For example, a large passenger vessel, 500 feet long, was torpedoed without warning and capsized and sank in 5 minutes. The explosion laid open two compartments to the sea, but this would not have been in itself sufficient to sink the ship. Unfortunately the watertight door leading into a third compartment was open at the time, so that actually three compartments were opened to the sea. The large area of waterplane lost on this account was enough to destroy the stability of the ship, making her capsize. Other cases occurred where damage was received in the after part of the ship, destroying a shaft tunnel and thus admitting water to the engine-room through the watertight door at the forward end of the tunnel which was left open.

In the case of a collision a bulkhead between two compartments may be damaged, and it is of the utmost importance that the inrush of water should be confined to the smallest limits. The suggestion was made by the Court of Inquiry into the *Empress of Ireland* disaster that in circumstances of danger it would be desirable to close all watertight doors and port holes below the top of the watertight bulkheads and to keep them closed until the danger was passed. It will be remembered that the *Empress of Ireland* was sunk in collision with a collier; many lives were lost. The advice given by the Court of Inquiry is undoubtedly sound in theory, but in practice—especially in peace time—since the closing of ordinary watertight doors causes such inconvenience in the ship, it is not likely to be taken. If it is logical to leave out watertight doors in a ship because they are dangerous, it is equally as logical to leave out port holes.

During the war, a committee appointed by the Institution of Naval Architects recommended that all existing watertight doors low down in main bulkheads should be secured so that they cannot be opened, and that if watertight doors be necessary they should be fitted high up in the bulkhead. This was considered by the committee to be particularly important when bunker coal is carried forward of the boiler-room bulkhead, as a watertight door through which coal is being trimmed cannot be depended upon. In the ordinary way this would mean lifting the coal in the reserve bunker over the top of the boiler-room bulkheads, or through the watertight door openings cut high up in the bulkhead, thus necessitating extra work for the crew of the ship and an increase in their number.

In spite of all that has been said for and against watertight doors, the fact remains that watertight doors are still fitted, the evidence pointing to the convenience of working the ship outweighing the possible danger brought about by fitting the doors. That this is so seems fairly obvious, since the necessity for closing a door in an emergency may come once in, say, ten years, but the desirability of having it there occurs several times a day.

Access must be provided to coal bunkers, and although every effort should be made in designing a ship to arrange that the bunker bulkheads are non-watertight, this is by no means always possible, particularly under the Subdivision Rules. Then, again, there must be access between the boiler rooms and from them to the engine-room. Shafting requires

periodical attention and watertight doors must be provided in the shaft tunnel for this purpose. It has been suggested that if watertight doors are dispensed with in the machinery spaces, access can be obtained to each watertight compartment from the top deck. This would mean increasing the number of engineer officers in a ship and adding to the wages bill. In addition, access trunks would have to be fitted in the 'tween decks or the engineers and greasers allowed in passenger accommodation. It is true that watertight doors are greatly cut down in warships, at least in the main watertight bulkheads, but in these vessels the number on the engine-room staff is not important, and in any case there are no passengers to consider. Economical considerations are relatively of little importance.

Even in warships, where efficiency is the first consideration, the number of watertight doors fitted is enormous. Some twenty years ago there was a sustained discussion with regard to this very question, the principal participants being the late Sir William White and the late Lord Charles Beresford. The former was in the position of having to put in watertight doors to meet the wishes of the senior naval officers controlling the design of warships. Lord Charles Beresford was anxious to reduce these, and while it is true he made some very drastic suggestions with regard to watertight doors low down in the main machinery bulkheads, he only proposed to do away with 19 out of a total of 208 in the battleship *Magnificent*. Since then some of his recommendations have been carried out, but even today in a first-class battleship there are nearly 200 watertight doors.

In passenger ships, watertight doors are generally necessary in one 'tween deck. It is curious in this connection, that the latest Subdivision Rules have actually added to the number of watertight doors required in a ship. It was usual before these rules came into force to stop the watertight bulkheads at a deck below the weather deck. Now, to get satisfactory subdivision, they must often be carried up to this latter deck. In the shelter-deck type of vessel, for instance, the main transverse bulkheads were only carried to the upper deck, and in consequence the passenger accommodation could be arranged in the shelter 'tween decks without introducing watertight doors in that space. Under the new Subdivision Rules it is frequently found necessary to carry the watertight bulkheads up to the shelter deck. In practice, as already indicated, it is necessary often to move from one compartment to another in the shelter 'tween decks, so that the bulkheads must be pierced and watertight doors fitted. These doors must be left open when the vessel is at sea. Watertight doors in a 'tween deck may be hinged or sliding, depending on the height of the deck above the water line. The danger of watertight doors in passenger 'tween decks being left open has often been demonstrated. When a damaged vessel has sunk to such a level as to bring the water into this 'tween deck over the damaged compartments, the water has promptly spread throughout the ship.

If there should be doubt still as to the need for watertight doors it should be removed by reference to the report of the Bulkheads Committee on the Subdivision of Merchant Ships. This report says, "The number of openings in watertight bulkheads shall be reduced to the minimum compatible with the design and proper working of the ship."

To summarize the position; watertight doors are objectionable principally because they are kept open and because in the ordinary way they require a great amount of effort and a good expenditure of time to open and close them. The necessity is for a means of opening and closing watertight doors easily. To do this the door must be operated by power. With such doors the convenience of the departments is met without endangering the safety of the ship. The principle of having watertight

doors always closed can be applied to all those which are operated by power in the lower part of the ship, as, for instance, those in the machinery spaces. This disposes of the objection that after a ship has received damage it is not possible to close the door, since it is already closed. On the other hand, many prefer to leave the doors in the machinery spaces open.

We have remarked that it is quite hopeless to rely upon direct human action locally, so that the necessity for power operation controlled from some central position becomes apparent. The great advantage of a central control from the bridge is that this control is under the ruling brains of the ship, *i.e.*, either the captain or the first officer, men who have been trained to act quickly and with presence of mind in an emergency and who, moreover, will actually see danger approaching in 99 cases out of 100, and will be able to throw over their door control lever before the actual moment of impact, thus forestalling the danger of distorted door frames to which reference has already been made. With regard to doors in passenger 'tween decks which cannot be kept closed without a great deal of inconvenience, it is safe to say that the 'tween deck bulkheads would not usually be distorted by an ordinary peace time accident unless the bulkhead was completely destroyed. In consequence it is possible to operate these doors, but that they should be closed quickly goes without saying, and what has been said for power-operated doors, centrally controlled, applies.

With regard to watertight doors in bunker bulkheads, the great objection to them is that they are likely to become partially blocked with coal. Here, again, the power-operated door has a distinct advantage in that the power can be made great enough to make the door cut through the coal. Nearly all the prominent steamship lines have fitted their passenger vessels with a centrally-controlled, power-operated watertight door system, and reports show that during the war it has been more difficult to sink these, than ships fitted with the ordinary watertight doors. With the power-operated doors there is often a certain amount of nervousness at first, amongst the men, with regard to their use, but in practice it is found that this is overcome when the men see that the local control is in their own hands and is quite sufficient to act against the central control, although the latter is made to close the door after the local action is taken off, this being usually done automatically. A power system of watertight doors does cost extra money, but this can be looked upon as an insurance premium.

The International Convention Rules for watertight doors, in vessels carrying more than 200 passengers, make it necessary to have, either doors which close by their own weight or by power pressure, and in any case operated from the bridge. Actually it is not often possible, in practice, to make all the watertight doors in the machinery spaces slide vertically so that they will close by their own weight. The result is that power-operated doors must be fitted, so that the Convention Rules do, in effect, require a power-operated, centrally-controlled system of watertight doors in passenger steamers.

In a passenger vessel, therefore, the choice remains between solid bulkheads and centrally-controlled, power-operated doors, and the advantages of doors are so obvious as compared with the inconvenient system of unpierced bulkheads that, in these days of high wages and short working hours, it follows that the moderate expense of installing an efficient power system would quickly be exceeded by the wages bill where solid bulkheads were fitted. It appears certain that all liners will in future have their bulkheads pierced for watertight doors, and that such doors will be centrally controlled and operated by power.—"Engineering."

GERMAN COASTAL SUBMARINE MINE-LAYERS.

(C Class).

By Dr. Eng. Franz Werner, of the Naval Board.

When in the autumn of 1914 Flanders had fallen into our hands, the necessity arose for having a mobile coast defense as well as a stationary one. For this purpose the war administration requested the speedy construction of small submarines which were to carry torpedoes and mines. The submarine commission fulfilled this order by designing two boats, one solely for torpedoes, the so-called *B-I* type (compare "Schiffbau" XX No. 18, pages 485 to 496), the other solely for mine-laying, the *C* type. Work was begun on the first of the C type (designated in this article as *C-I*) in November, 1914. The Vulcan Works at Hamburg was commissioned to build ten and the Weser Stock Company five.

C-I TYPE.

The plans had been made by the construction department of the submarine commission. In the following only the characteristic distinctions will be taken up, as the properties peculiar to the coastal submarine, and the conditions governing its construction have been explained in the above-mentioned article about the *B* type, which also holds good for the mine-layer. The principal measurements are shown in Tables I to V. Table I (combination drawing) shows the outward form. The boats are especially remarkable for three peculiarities which exercised a lasting influence on the design.

FORM.

Especially noticeable with regard to the form is the ungainly cylindrical shape of the fore part which reminds one of a Zeppelin airship. There were two reasons which led to the choice of this peculiar shape. The first one was the space required for the mine-tubes in which the mines were stowed. The second was the necessity of transporting the boats to Flanders by rail. This led to the reduction of the maximum diameter to a size which would be suitable to the dimensions permitted by railroad cars, tunnels, bridges, etc. The shape of the special railroad cars necessitated the outlines of the fore part of the vessel.

CONNING TOWER.

The second characteristic of the boats is the high conning tower. All the foreign submarines of the same size were furnished with observation hoods, and not with real conning towers, consequently the commander's post is only slightly above the water when traveling on the surface. The submarine commission knew very well that these vessels would only be able to maintain themselves if they could travel on the surface for considerable distances. This, however, necessitates a sufficient sea-worthiness. The latter can only be insured if the conning tower is tall enough to give protection from the sea to commander and helmsman. The natural outcome of this was the furnishing of the commander's post with a pressure-resisting substructure tall enough for at least one man to stand upright in.

TABLE 1

CHIEF MEASUREMENTS AND QUALITIES

	UC-I	UC-II	UC-III
Length over all (m)	34.0	41.85	55.10
Maximum beam (m)	3.13	3.20	3.55
Depth of keel with maximum oil supply on board (m)	3.04	3.645	3.768
Height of keel (m)	.5	.56	.56
Height of keel (lower section) with upper edge O.D. measured for ½ length (m)	3.2	4.6	4.7
Height to top of periscope (m)	6.3	7.6	7.7
Surface displacement with			
(a) normal oil supply (cbm)	177.0	417.0	480.0
(b) maximum " " "	177.0	455.0	491.6
Displacement submerged			
(a) without free flooding spaces (cbm)	192.5	508.6	559.27
(b) total form displacement (cbm)	225.0	550.0	716.0
Capacity of ballast tanks including additional bunker (cbm)	14.6	82.02	91.0
Capacity of trimming tanks (cbm)	5.0	20.3	24.1
Capacity of ordinary bunker (cbm)	2.5	46.6	65.6
Cap. of additional bunker (cbm)		16.6	12.5
Capacity of lubricating oil tanks (cbm)	0.52	5.67	4.5
Surface engines H.P.	90	500-500	600-650
Maximum surface speed, knots per hour	6.5	11-12	11-12
Surface cruising radius, nautical miles	800 at 5.5 kn.	10000 at 7 kn.	8000 at 8 kn.
Under water engines H.P.	136.0	460.0	600.0
Number of cells and type of accumulator battery	2 x 56 MAS	2 x 62 MAS	2 x 62 MAS
Capacity of the accumulator battery for 10 hours in ampere hours	2700.0	4800.0	5900.0
Maximum speed submerged knots per hour	5.0	7.0	6.5
Maximum cruising radius, submerged, at 4 knots per hr., nautical miles		55.0	56.0
At 5 kts. per hr., nautical miles		92.0	95.0
Torpedo equipment (maximum number carried)	-	2 tubes for'd	2 amidships
	-	1 tube aft	1 aft
	-	7 torpedoes 50 cm	7 torpedoes 50 cm
Mines	12.0	18	14
Guns	1 M.G.	1-8.8 cm (100)	1-10.5 cm (170)
		1 M.G.	1 M.G.
Number of crew	18	28	32
Admissible depth for diving (m)	50	50	75
Time of diving to a depth of 9 m. (sec)	50	40	45
Total cost of construction in marks	700,000	1,700,000	3,000,000

TABLE II

STABILITY MEASUREMENTS

	UC-I	UC-II	UC-III
1. Stability on surface			
F (Center of displacement) above lower edge of pressure hull (m)	1.45	1.722	1.865
M (Metacentric width) above F (m)	.07	.557	.615
G(Center of gravity) above lower edge of pressure hull (m)	1.27	1.804	1.949
Metacentric height MG	0.25	0.455	0.529
2. Stability under water for γ= 1.0			
F above lower edge of pressure hull	1.59	1.986	2.043
G above lower edge of pressure hull	1.84	1.804	1.909
FG (length & Breadth)	0.35	0.182	0.134

TABLE III

DATA CONCERNING PRESSURE HULLS AND OTHER IMPORTANT PARTS

	UC-I	UC-II	UC-III
Length of pressure hulls (m)	39.52	40.40	43.0
Max. Dia. " " (m)	5.15	5.652	5.652
Thickness of plates of pressure hulls (mm)	10	11	11
	Mine space 11 on side 14 above and below	Mine space 11on side 12 above 13 below	Mine space 11on side 12 above 13 below
Frame profile (mm)	∠ 130 x 65 x 8 (amidships) ∠ 150 x 70 x 10 (mine space)	∠ 150 x 70 x 10	∠150 x 70 x 8.5
Distance between frames (mm)	750	800	800
Total weight of pressure hull (Sides, front.. (kg)	55470	55800	62655
Total volume of pressure hull (cbm)	185.5	314	358.6
Weight of pressure hull per cbm of vol. kg/cbm	19.36	17.8	18.5
Measurements of conning Tower (mm)	1250	1250	1250
Thickness of top of conning tower (mm)	15	12	12
Plate thickness of outer hull (mm)	-	3.5-4	5
Frame profile of outer hull (mm)	-	∠ 60x40x5	∠ 60x40x5 ∠ 50x40x5
Distance between frames of outer hull (mm)	-	4-800	800
Plate thickness of spaces open to the sea (mm)	1-2	2	2

TABLE IV

	UC-I	UC-II	UC-III
Main rudder (sqm)	1 x 1.75	2 x 2.15	2 x 2.15
Forward rudder (sqm)	2 x 1.0	2 x 1.75	2 x 1.75
After rudder (sqm)	1 x 2.0	2 x 1.6	2 x 1.6

ARRANGEMENT FOR THE MINES.

The third peculiarity is the arrangement for the mines. In the fore part of the boats are six slanting tubes which pass completely through the pressure hull so that, when the vessel is submerged, the tubes are entirely filled with water. In each tube two mines are stowed one above the other. See drawing 5. The cams reach under the projections *b* of the mine anchors and thus hold them firmly in place in the tube. The cams may be screwed into the openings *c* of the mine tube from the interior of the boat whereby the mine is deprived of its support and drops downwards. The mine had to undergo special improvements in order to adapt it to the dropping contrivance. Our mines up to that time were not adapted to this arrangement. But it was necessary to avoid changing the lead ignition cap which had already successfully stood the test of use. The ignition cap of the mine was now considerably endangered by the walls of the tube particularly during the fall so that special contrivances had to

TABLE V.

WEIGHTS

	UC-I		UC-II		UC-III	
	t	%	t	%	t	%
Hulls	81.25	46.0	182.1	45.00	205.0	42.8
Engines	40.50	22.9	111.3	26.00	144.6	30
Torpedo equipment	-	-	14.7	3.5	14.1	3.0
Gun equipment	-	-	4.2	1.0	10.7	2.3
Mine equipment	14.0	7.9	14.67	3.5	12.1	2.5
Fixtures			6.8	1.6	5.5	1.1
Materials and lub. oil			5.8	1.4	6.6	1.4
Crew, provisions	15.6	8.7				
Water			11.7	2.75	12.2	2.6
Fuel, normal			41.0	9.25	56.8	11.6
Ballast and reserves	25.75	14.5	24.7	6.0	13.3	2.7
	177.0	100.0	417.0	100.0	480.0	100.0

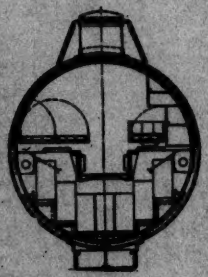
DISPLACEMENT

	UC-I (cbm)	UC-II (cbm)	UC-III (cbm)
Volume of pressure hull	169.0	313.98	338.5
Outer hull, volume (Pressure proof cells and additions)	9.0	57.4	78.9
Displacing fuel		46.6	63.6
Total submerged displacement without ballast cells	178.0	418.0	481.0
Residue	1.0	1.0	1.0

be made to ensure the safe exit of the mines from the U-boats. For this purpose hinged guide-rails were attached to the mine-anchor in such a way as to give the mine the correct direction while leaving the tube, thus protecting the lead ignition caps from contact with the walls of the tube or with the vessel. (See drawing 5.) After the mine has dropped down and has settled on the ocean-bottom, these guide-rails fold back and lie on the ocean bottom like grippers, in this way permitting the mine to float freely upwards and the anchor cable to unwind without hindrance.

GENERAL ARRANGEMENT PLAN FOR UC-1 BOATS

TRIMMTANK	TRIMMING TANK
SCHALTTAFEL	SWITCHBOARD
OELBUNKER	OIL FUEL TANK
AKKUMULATOREN	ACCUMULATORS
TAUCHTANK	BALLAST TANK
MINENAUSGLEICHTANK	MINE COMPENSATING TANK
FRISCHWASSERTANK	FRESH WATER TANK



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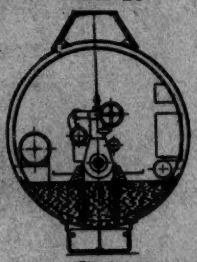

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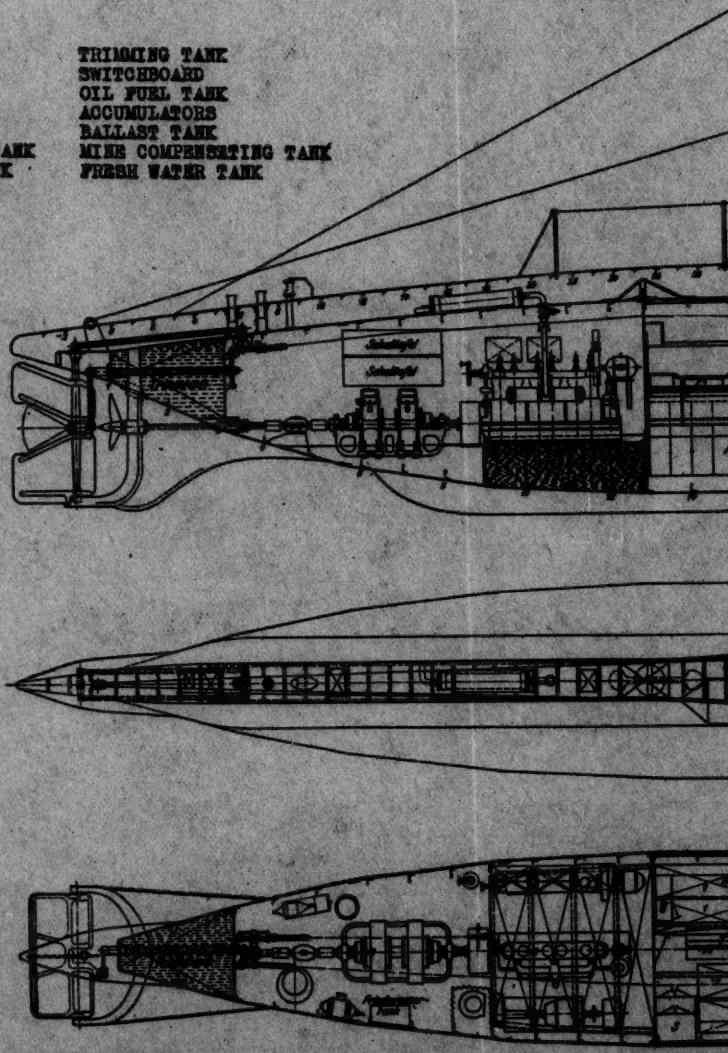

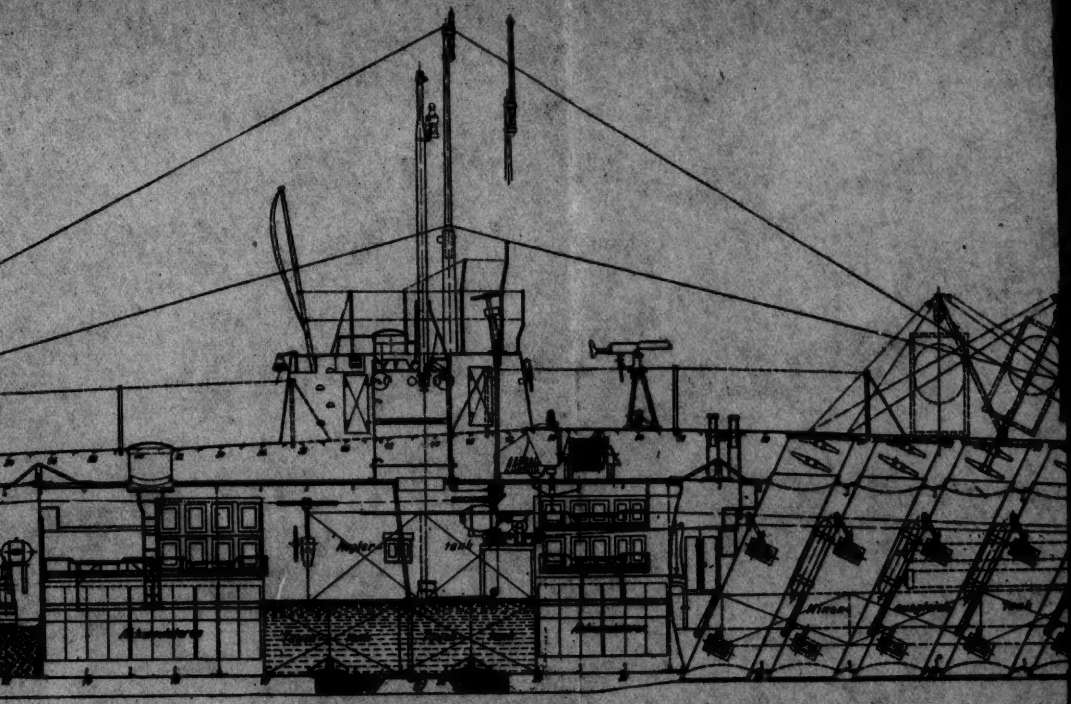

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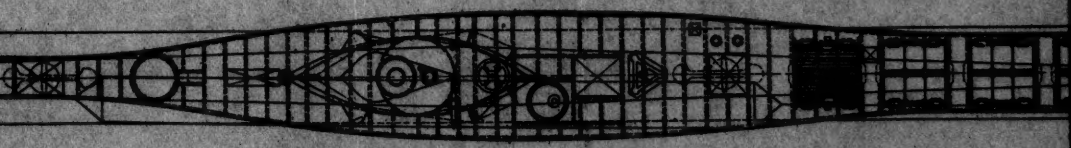
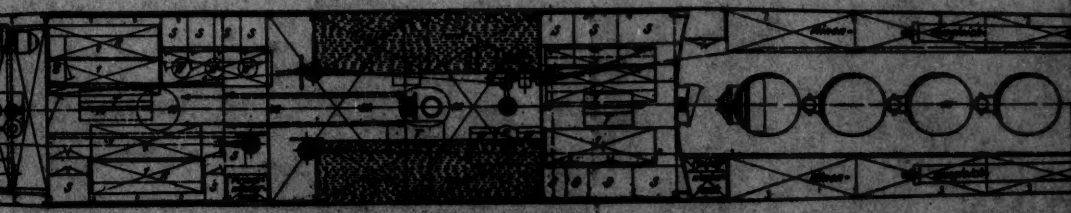
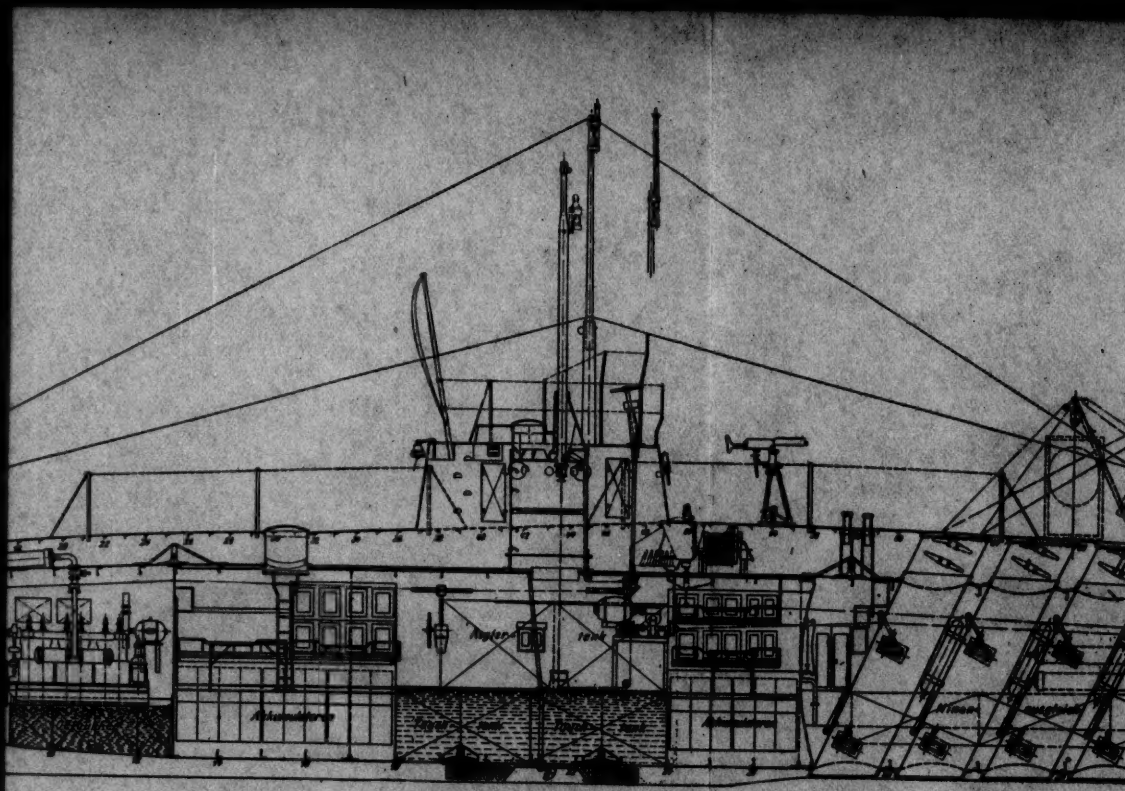


DECK PLAN



INTERIOR ARRANGEMENT PLAN



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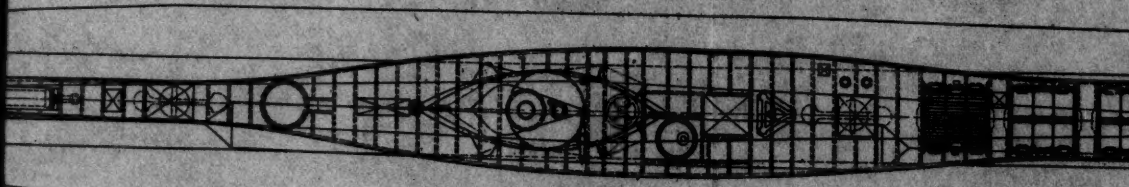
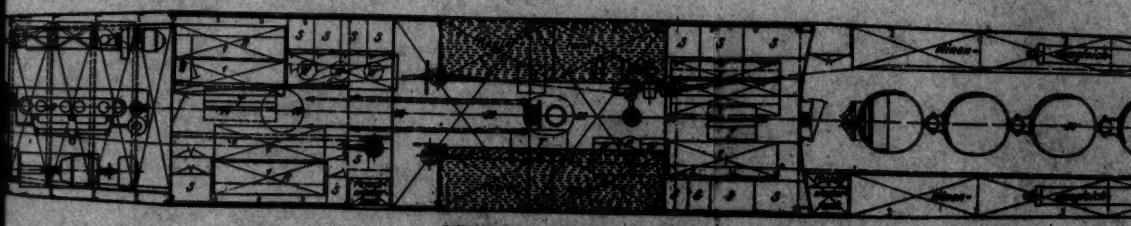
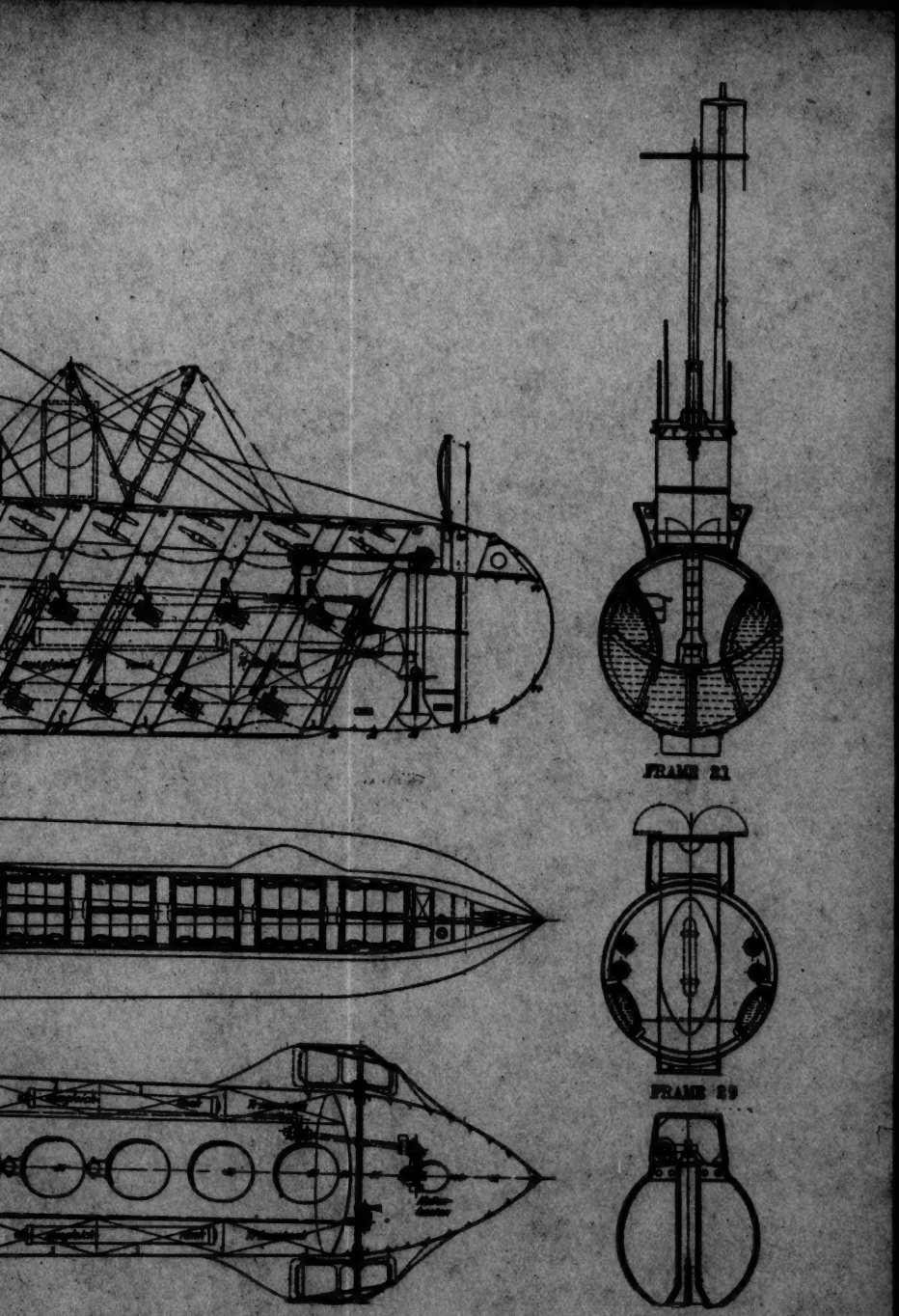


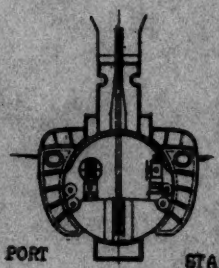
DECK PLAN



INTERIOR ARRANGEMENT PLAN


FRAME 21
FRAME 29

1. AIR FLASKS
2. EMERGENCY RUDDER
3. WAR HEADS
4. CONVERTER FOR GYROSCOPIC COMPASS
5. EXHAUST FUNNEL
6. CONVERTER FOR RADIO INSTALLATION
7. AUXILIARY AIR COMPRESSOR
8. PRINCIPAL AIR COMPRESSOR
9. AIR COOLER
10. CONVERTER FOR CENTRAL COMMUNICATIONS STATION
11. MAIN ELECTRIC MOTOR
12. STARTING BOTTLES
13. SWITCHBOARDS



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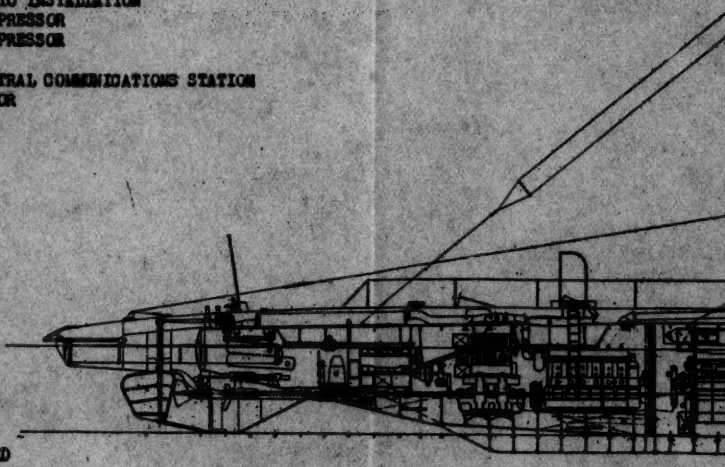

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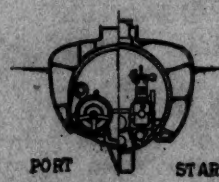

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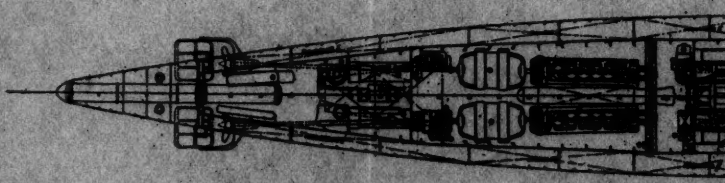

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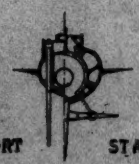
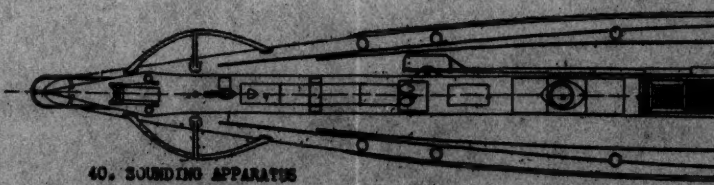


OUTER-FRAME 7 INNER-FRAME 1



40. SOUNDING APPARATUS
41. ANCHOR GEAR
42. COMPASS
43. RUDDER
44. EMERGENCY PERISCOPE
45. MAIN PERISCOPE
46. MACHINE GUN

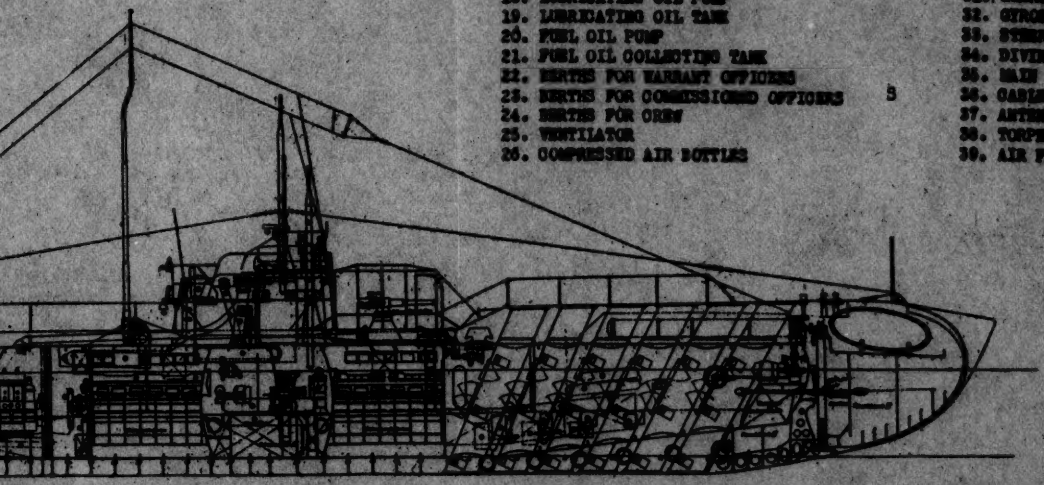
[illegible] & Ltd. Asst. to Engine Constructors Barrow in Furness

NERAL ARRANGEMENT PLAN FOR UC-II

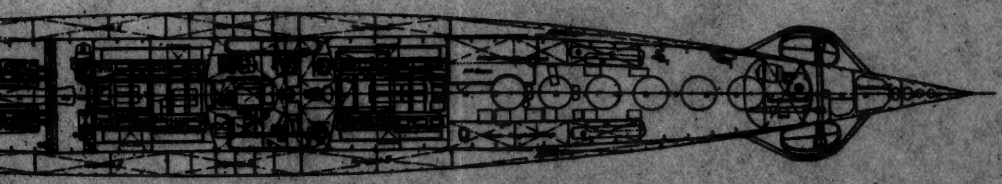
1:150

14. MAIN OIL MACHINE
15. VESSEL FOR FUEL SPRAYING
16. OIL COOLER
17. PRINCIPAL BILGE PUMP
18. LUBRICATING OIL PUMP
19. LUBRICATING OIL TANK
20. FUEL OIL PUMP
21. FUEL OIL COLLECTING TANK
22. BERTHS FOR WARRANT OFFICERS
23. BERTHS FOR COMMISSIONED OFFICERS
24. BERTHS FOR CREW
25. VENTILATOR
26. COMPRESSED AIR BOTTLES

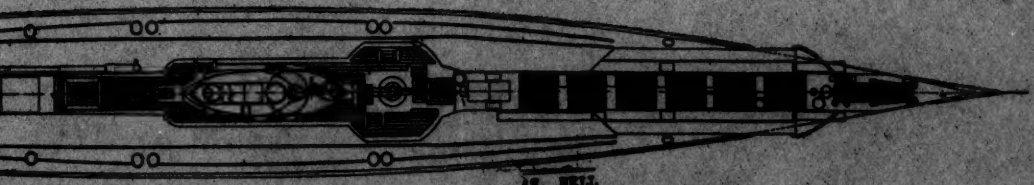
27. TRIMMI
28. TURBO B
29. MANOEUV
30. STARTER
31. MANOEUV
32. GYROSCO
33. STEERIN
34. DIVING
35. MAIN PU
36. CABLE R
37. ANTENNA
38. TORPEDO
39. AIR FLA



LONGITUDINAL SECTION



INTERIOR ARRANGEMENT PLAN



DECK PLAN

47. BELL
48. LIGHT BUOY
49. AIR EXHAUST
50. FRESH AIR INTAKE
51. BLOWING OUT VALVE FOR BALLAST TANK II and III
52. STERN LIGHT

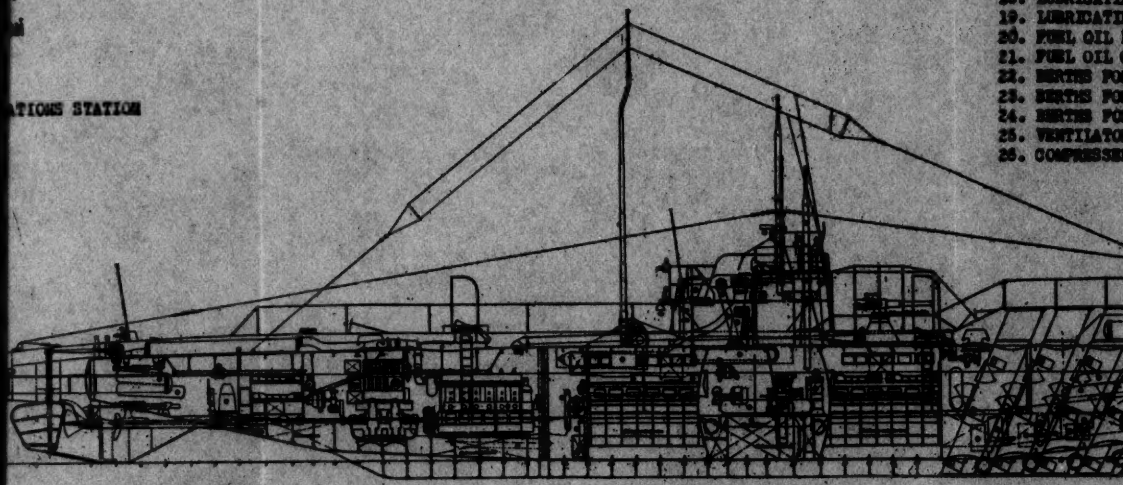
INDE

GENERAL ARRANGEMENT PLAN FOR UC-II

1:150

...ATIONS STATION

14. MAIN OIL
15. VESSEL FO
16. OIL COOLE
17. PRINCIPAL
18. LUBRICATI
19. LUBRICATI
20. FUEL OIL
21. FUEL OIL
22. BERTHS FO
23. BERTHS FO
24. BERTHS FO
25. VENTILATO
26. COMPRESSE



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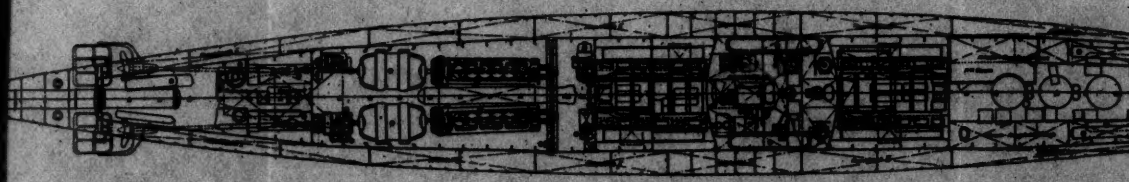
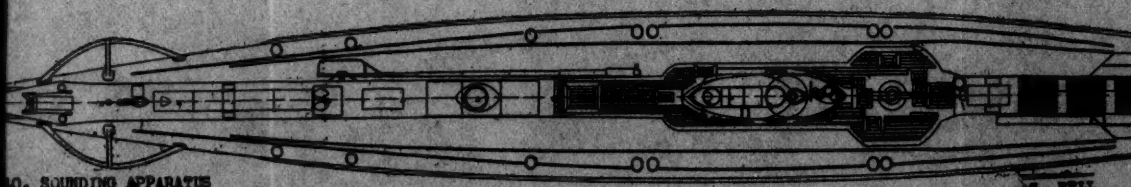


INTERIOR ARRANGEMENT PLAN



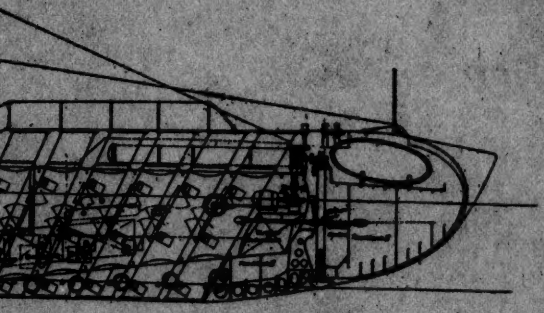
...0. SOUNDING APPARATUS
...1. ANCHOR GEAR
...2. COMPASS
...3. RUDDER
...4. EMERGENCY PERISCOPE
...5. MAIN PERISCOPE
...6. MACHINE GUN

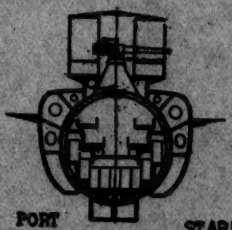
DECK PLAN

47. BELL
48. LIGHT B
49. AIR EXH
50. FRESH A
51. BLOWING
52. STERN L

TABLE II

14. MAIN OIL MACHINE
15. VESSEL FOR FUEL SPRAYING
16. OIL COOLER
17. PRINCIPAL BILGE PUMP
18. LUBRICATING OIL PUMP
19. LUBRICATING OIL TANK
20. FUEL OIL PUMP
21. FUEL OIL COLLECTING TANK
22. BERTHS FOR WARRANT OFFICERS
23. BERTHS FOR COMMISSIONED OFFICERS 5
24. BERTHS FOR CREW
25. VENTILATOR
26. COMPRESSED AIR BOTTLES
27. TRIMMING PUMP
28. TURBO BLOWER
29. EMERGENCY TOILET
30. STARTER FOR TURBO BLOWER
31. EMERGENCY PERISCOPE WINCH
32. GYROSCOPIC COMPASS
33. STEERING WHEEL
34. DIVING RUDDER MOTOR
35. MAIN PERISCOPE WINCH
36. CABLE DRUM FOR WIRELESS MAST
37. ANTENNAE MAST WINCH
38. TORPEDO MUNITIONS LOCKER
39. AIR FLASKS





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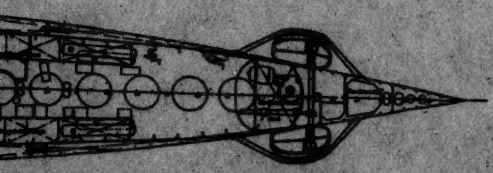
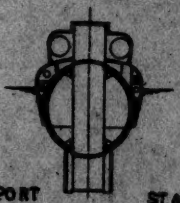
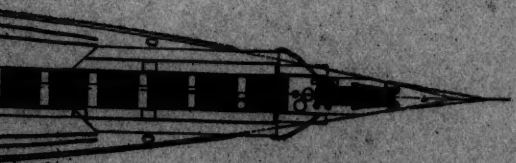
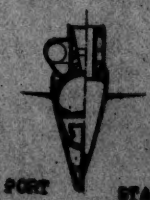




INNER-FRAME 43



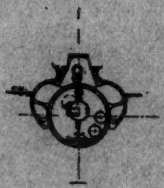


INNER-FRAME 46 OUTER-FRAME 94

47. BELL
48. LIGHT BUOY
49. AIR EXHAUST
50. FRESH AIR INTAKE
51. BLOWING OUT VALVE FOR BALLAST TANK II and III
52. STERN LIGHT

Querschnitte

FRAME 7



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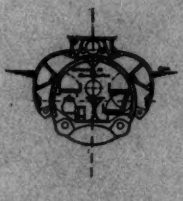

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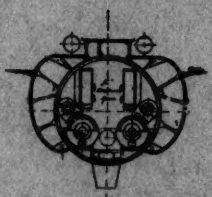

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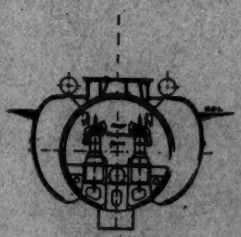

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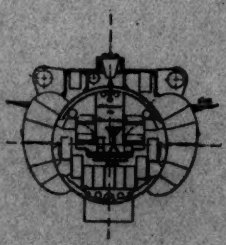

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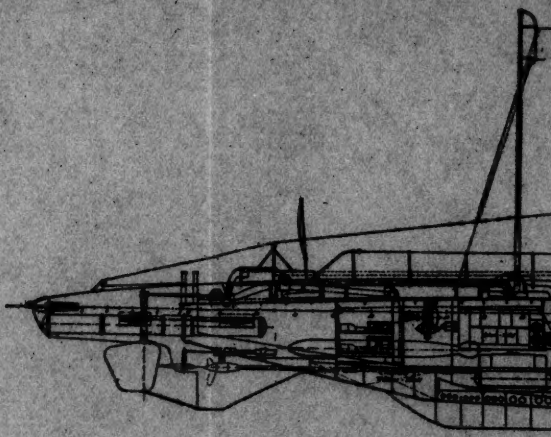
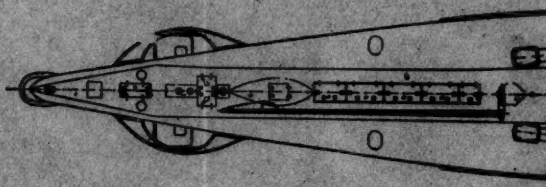
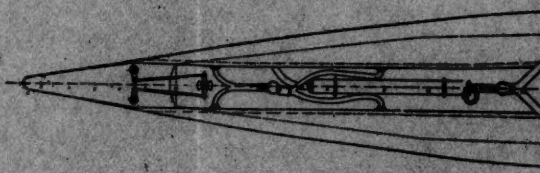
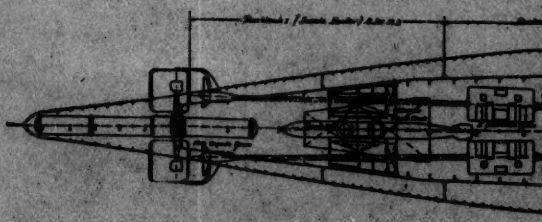


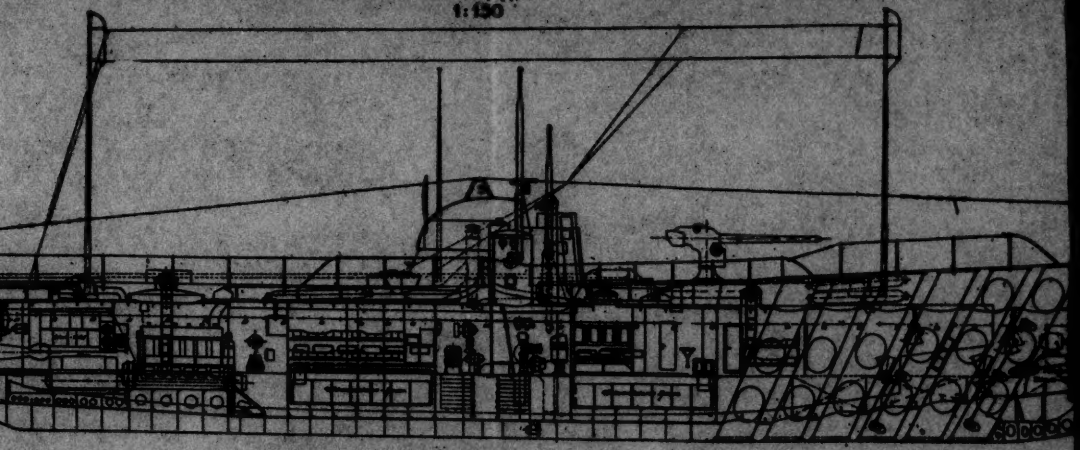




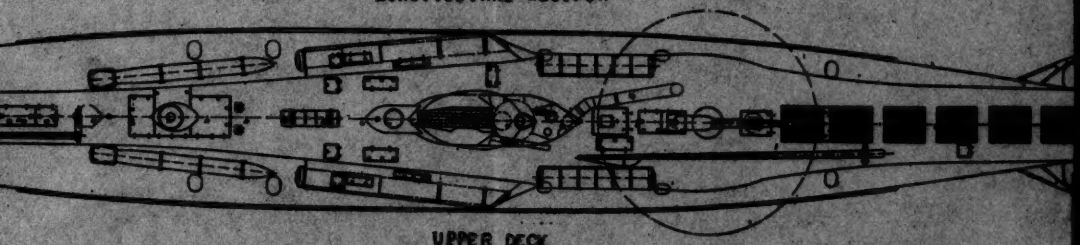


GENERAL ARRANGEMENT PLAN FOR DC-III

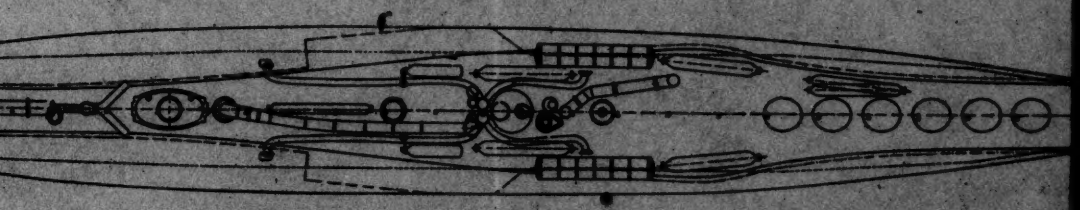
1:150



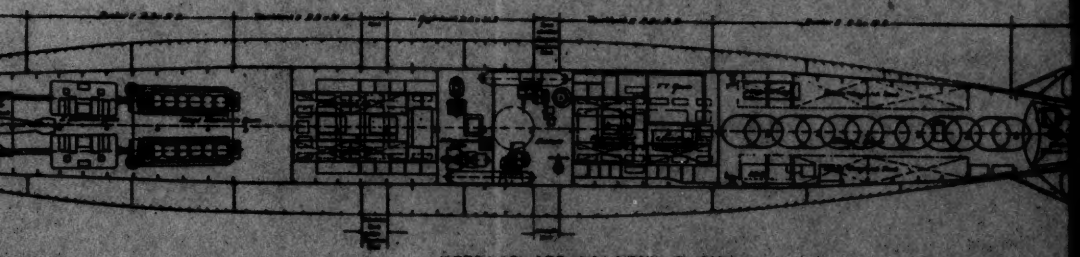
LONGITUDINAL SECTION



UPPER DECK



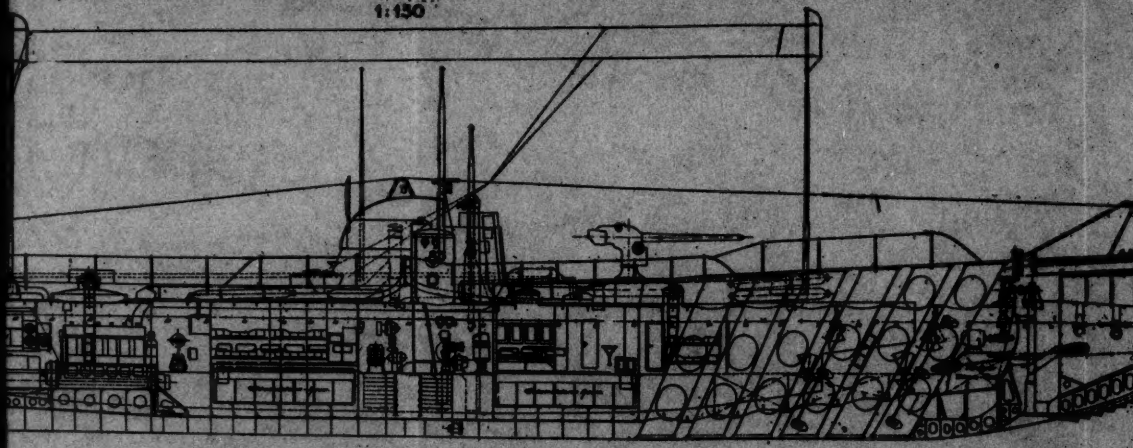
DECK PLAN



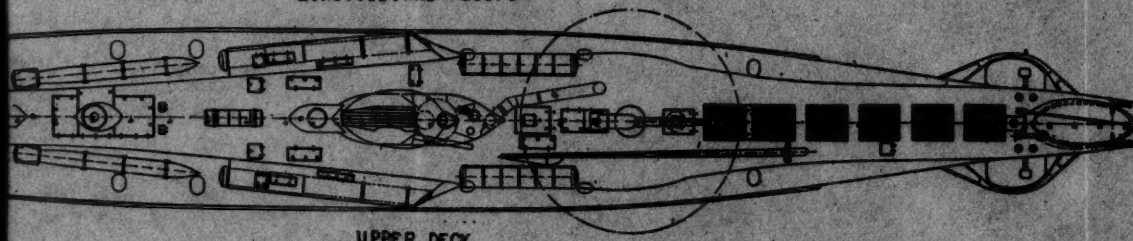
INTERIOR ARRANGEMENT PLAN

GENERAL ARRANGEMENT PLAN FOR DC-III

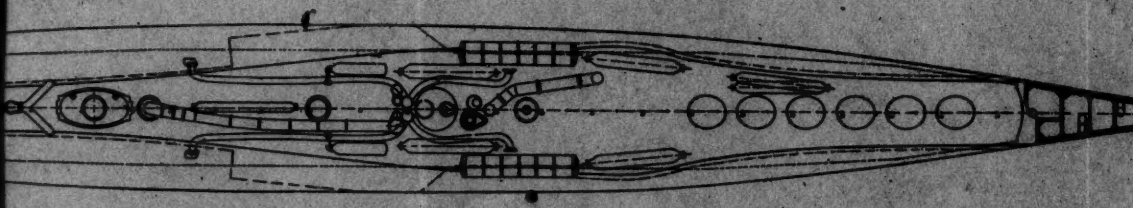
1:150



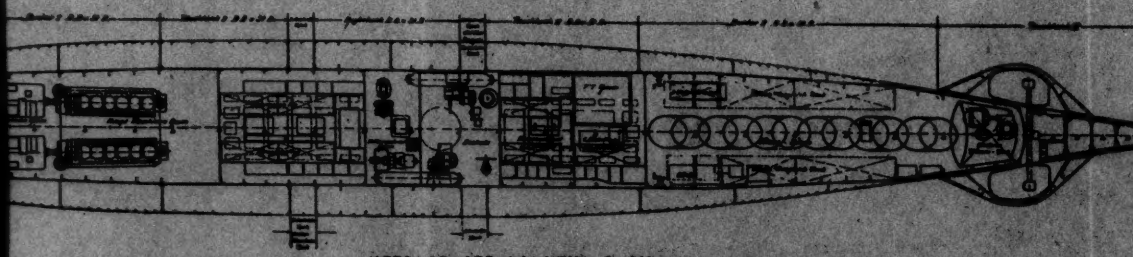
LONGITUDINAL SECTION



UPPER DECK

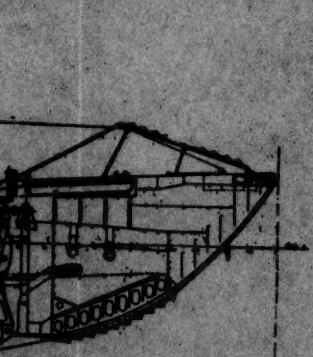


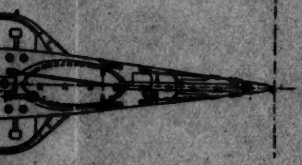
DECK PLAN

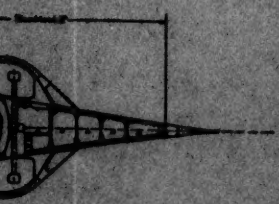


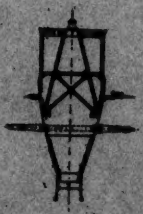
INTERIOR ARRANGEMENT PLAN

TABLE III



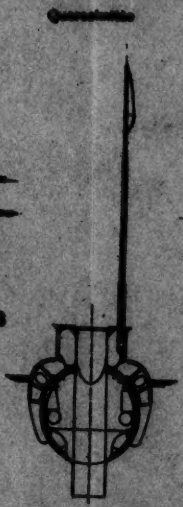




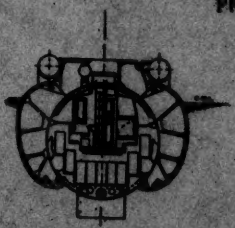


FRAME 88

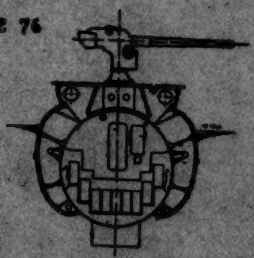
FRAME 98



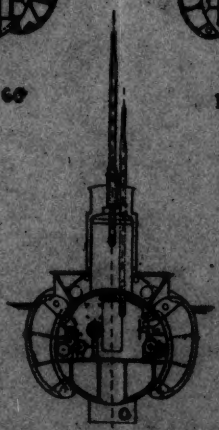
FRAME 76



FRAME 60



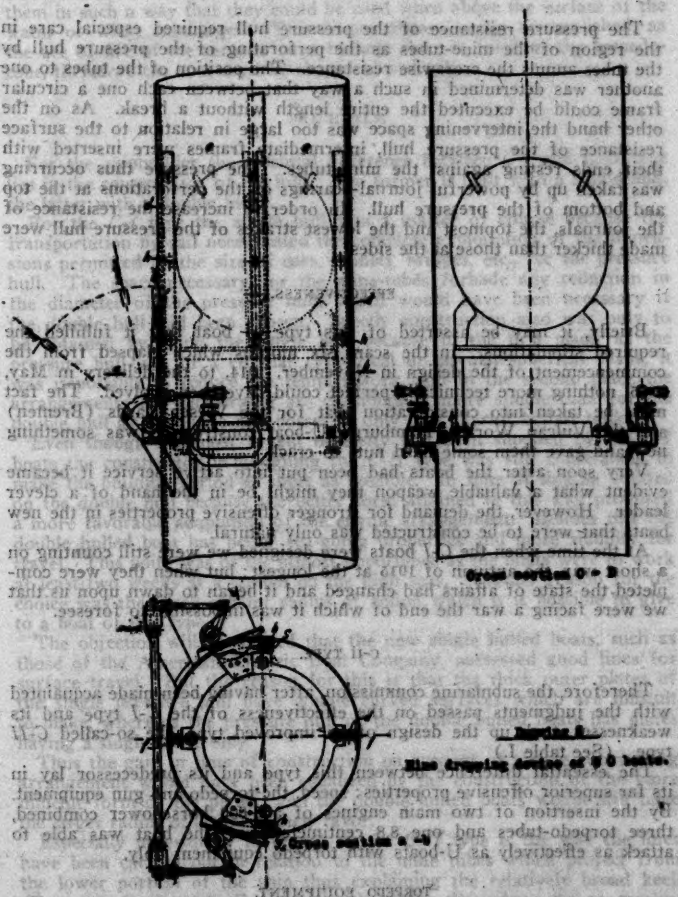
FRAME 66



FRAME 54

THE MINE-TUBES.

It was necessary to incline the mine-tubes so that the mines might be released while the boat was in motion. The degree of inclination was determined by experiments with models. The experiments were executed on the basis of the highest attainable water-velocity although the mines are



usually released at considerably lower rates of speed. It was, however, observed that an inclination which was greater than the respective speed of the vessel's demands is not detrimental to the falling of the mine.

The mines are guided during their exit by flat iron rails *f*, two attached lengthwise and two crosswise in the angles *g* arranged in the cross-section of the tube. For this purpose the above-mentioned guiding-rails of the mines are furnished with guiding-rollers *h*. The axes of the latter are alternately radial and tangential to the mine.

PRESSURE HULL.

The pressure resistance of the pressure hull required especial care in the region of the mine-tubes as the perforating of the pressure hull by the tubes annuls the crosswise resistance. The position of the tubes to one another was determined in such a way that between each one a circular frame could be executed the entire length without a break. As on the other hand the intervening space was too large in relation to the surface resistance of the pressure hull, intermediate frames were inserted with their ends resting against the mine-tubes. The pressure thus occurring was taken up by powerful journal-bearings on the perforations at the top and bottom of the pressure hull. In order to increase the resistance of the journals, the topmost and the lowest strakes of the pressure hull were made thicker than those at the sides.

EFFECTIVENESS.

Briefly, it may be asserted of this type of boat that it fulfilled the required stipulations. In the scant six months which elapsed from the commencement of the design in November, 1914, to the delivery in May, 1915, nothing more technically perfect could have been evolved. The fact must be taken into consideration that for the Weser Yards (Bremen) and the Vulcan Works (Hamburg) U-boat construction was something new and gave them some hard nuts to crack.

Very soon after the boats had been put into active service it became evident what a valuable weapon they might be in the hand of a clever leader. However, the demand for stronger offensive properties in the new boats that were to be constructed was only natural.

At the time when the *C-I* boats were designed we were still counting on a short war, the autumn of 1915 at the longest; but when they were completed the state of affairs had changed and it began to dawn upon us that we were facing a war the end of which it was impossible to foresee.

C-II TYPE.

Therefore, the submarine commission, after having been made acquainted with the judgments passed on the effectiveness of the *C-I* type and its weaknesses, took up the design of an improved type, the so-called *C-II* type. (See table I.)

The essential difference between this type and its predecessor lay in its far superior offensive properties: speed, the torpedo and gun equipment. By the insertion of two main engines of 500-600 horsepower combined, three torpedo-tubes and one 8.8 centimeter gun, the boat was able to attack as effectively as U-boats with torpedo equipment only.

TORPEDO EQUIPMENT.

As the mines occupied the entire space in the pressure hull the only remaining position for the torpedo-tubes was on the outside of the boat.

In order to be able to retain accuracy of aim it was decided to choose pressure-proof, water-tight outside tubes and to dispense with the use of the so-called grating or despatch tube used by the French and the Russians.

The place adjacent to those portions of the mine-tubes which projected above the pressure hull turned out to be the suitable location. Here the tubes could be placed without disturbing the outer lines parallel to the axis of the boat to any considerable degree. It was also possible to place them in such a way that they could be used when above the surface of the water or when submerged. This is not always easily accomplished, as the arc of fire of the torpedo must be kept clear in front of the torpedo-tube, therefore the boat must be shaped accordingly.

THE HULLS.

For the shipbuilder the essential difference between *C-I* and *C-II* type lies in the construction of the hull. The former is built with a single hull, the latter with a double hull.

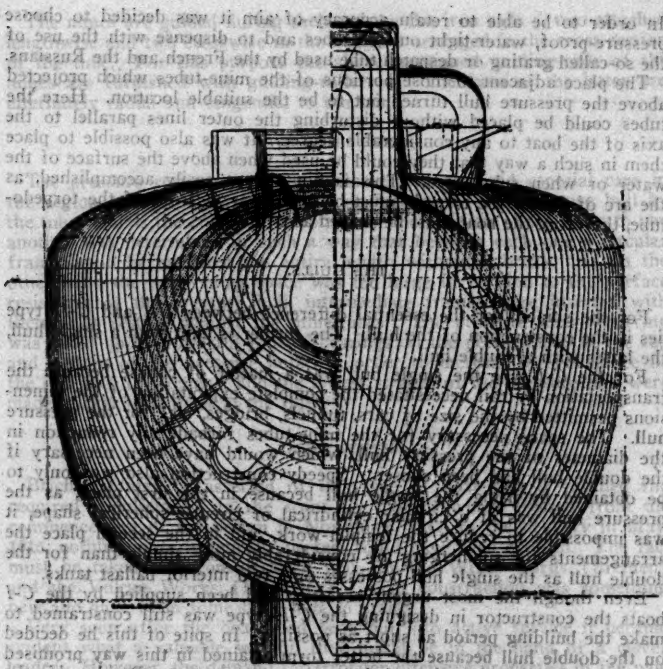
For the *C-I* type the single hull was a matter of course because the transportation by rail necessitated the complete exploitation of the dimensions permitted by the size of cars, tunnels, bridges, etc., for the pressure hull. The space necessary for the mine-tubes forbade any reduction in the diameter of the pressure hull, which would have been necessary if the double hull had been chosen. Speedy construction also was only to be obtained by using the single hull because in the first place, as the pressure hull was of a circular cylindrical or circular-spherical shape, it was impossible to reduce the templet-work, and in the second place the arrangements for submerging are considerably more simple than for the double hull as the single hull possesses only two interior ballast tanks.

Even though the most urgent necessity had been supplied by the *C-I* boats the constructor in designing the *C-II* type was still constrained to make the building period as short as possible. In spite of this he decided on the double hull because the better form attained in this way promised a more favorable adaptation of the engine arrangement. Besides this the double hulled boat has better sea-going properties and more stability for travel. This point was of the utmost importance as the boats were to work at least two weeks at a stretch on the high seas. Another reason for the choice was the much greater radius of action which it is possible to give to a boat of this type.

The objection will be offered that the new single hulled boats, such as those of the American Electric Boat Company, possessed good lines for surface travel. But the reason for this is that the thick outer plates of the single pressure hull have double curved surfaces which are difficult to make, where as the pressure hull of the double hulled boats has plates having a single curve only.

Thus the gain in time of construction on account of simple submerging arrangement would have been equalized by the construction of the more difficult form of the plates; and the double hulled boats would still have the larger radius of action in their favor.

Especially noteworthy in regard to the *C-II* type is the way the lines have been carried out in relation to the mine tubes which project from the lower portion of the ship thus explaining the relatively broad keel. (See reproduction 8.) For the rest they have throughout the appearance of a normal surface vessel.



Reproduction of U II type construction plan.

EFFICIENCY.

Boats of this type have stood the test of service well; sea-worthiness, submerging qualities, its action when submerged and the manœuvering capabilities gave perfect satisfaction. The obtained speed of 11½ to 12 knots was, in view of the strong offensive qualities, relatively high; the radius of action in spite of the small displacement was so extensive that the boats could be employed all round the British Isles, in the Atlantic as far as the coast of Morocco, and could be sent from Kiel to Pola without renewing the fuel supply.

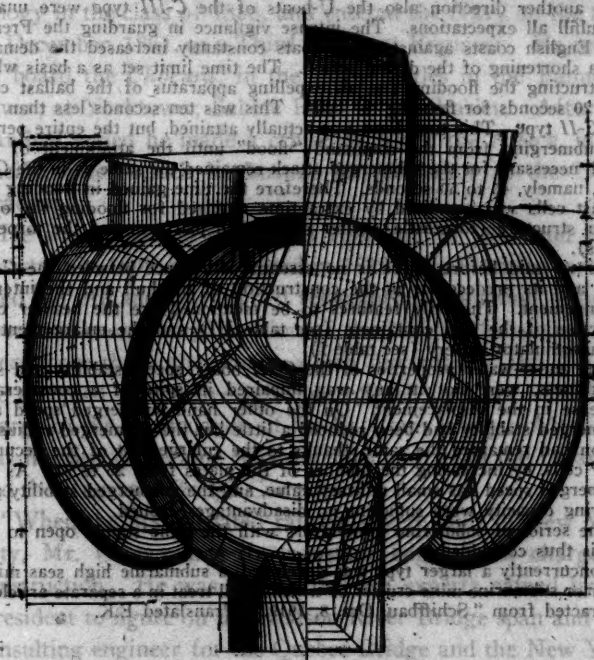
But they possessed two features which caused the chiefs to instigate a change in the construction of the *U-I* when at a later date, mine layers were ordered in considerable numbers. These features were the wet bridge which was a consequence of the speed combined with the high forecastle and the position of the torpedo-tubes at the side; and the easily flooded gun platform which was caused by the well occurring between forecastle and conning tower. Both of these were military disadvantages not to be underestimated. They rendered the sighting and recognizing of the adversary very much more difficult and impaired accuracy when aiming the weapon.

A submarine more than any other war-vessel is never a thing of perfection but will always remain a compromise, with defects of more or less importance. Therefore such defects as those mentioned above, although in themselves small, stand out so prominently for the reason that they interfere with the daily life of the crew and they are therefore often subject to more severe and lasting criticism than more serious and fundamental defects such as lack of speed, stability, etc.

C-III TYPE.

When, therefore, after a time of less necessity for mine-layers, there arose again in 1917 the demand for a great number, the Submarine Commission decided to design a slightly different type, but which only deviated outwardly from the *C-II* type, the interior remaining practically unchanged. The details of the plans were prepared at the wharf of Blohm and Voss and the Danzig Imperial wharves, who alone constructed or prepared the boats of this type with the exception of three which were built by the Weser yards; Blohm and Voss having the construction of 89 to their credit.

The new form shown in table III was the result of the effort to avoid the defects mentioned. (See reproduction 9.) It is characterized by the



Reproduction 9 C III type Construction plan.

gently sloping strake of the forecastle to the gun platform. The position of the gun itself is, however, more elevated and the torpedo-tubes which were formerly outside in the bows and were said to cause the spray were placed much farther back, so that the mouth pieces were abreast of the conning tower.

It was evident that these compromises also possessed disadvantages. Even while making the design, the constructor found it most convenient for the torpedo tubes to be no longer placed parallel with the lengthwise axis of the hull, thus ensuring a free field for the torpedo projectile, but at an angle which deviated several degrees from the above. The result was a very unfavorable shape for submerged navigation. The new forecastle, which took in the gun platform, increased the submerged displacement considerably. Above all, the torpedo tube greatly enlarged the maximum midship frame when submerged. (See reproduction 9.) The result of the trial trips was a not inconsiderable decrease of the highest speed.

The new position of the torpedo tubes had done away with the disagreeable spraying of the bridge, but in place of this, when in a sea of four points, brightly shining lines of foam were formed at the caps of the torpedo tubes, thus increasing the visibility of the vessel when in motion. This again was a military disadvantage.

In another direction also the U-boats of the *C-III* type were unable to fulfill all expectations. The intense vigilance in guarding the French and English coasts against the U-boats constantly increased the demand for a shortening of the diving period. The time limit set as a basis when constructing the flooding and air-expelling apparatus of the ballast cells was 20 seconds for flooding the cells. This was ten seconds less than for the *C-II* type. This time limit was actually attained, but the entire period of submerging from the command "flood" until the attainment of the depth necessary for the submerged attack remained the same as in the *C-II* type, namely, 45 to 50 seconds. Therefore the time gained in flooding the ballast cells was equalized by extra time required for flooding the open super structure (forecastle, shelter of the gun platform and the torpedo-tubes).

As we said before, there is no essential difference between the *C-III* type and its predecessor in the construction of the hull and the interior arrangement. The only features to be mentioned are the heavier construction of the fuel containers (see table III) and the enlargement of the accumulator battery (see table I).

The under-water properties of the *C-III* boats, speed, stability and seaworthiness, were not in any way impaired in spite of a considerable increase in the displacement. On the other hand submerged speed and submerged stability had been reduced a little but the submerged radius of action had remained the same, thanks to the enlargement of the accumulator cells, by increasing the number of the plates from 26 to 32. As the submerged speed has small military value, and the submerged stability and steering capacity were sufficient no disadvantages ensued.

The series of submarine mine-layers with the mine supply open to the sea is thus concluded.

Concurrently a larger type was developed, a submarine high seas mine-layer or submarine mine-cruiser of which I will treat in a separate article.—Extracted from "Schiffbau, Oct. 8, 1919." Translated E.K.

OBITUARY.

THEODORE COOPER.

Old officers, and particularly the older Engineers of the Navy, and the students at the Naval Academy from 1866 to 1869, will remember Theodore Cooper, who was, in his day, one of the brightest and most genial of the Engineers.

He was born in Coopers Plains, N. Y., eighty-one years ago and died in New York City during August last.

He was graduated at the Van Rensalaer Institute of Technology in the class of 1858, and entered the Navy on the 24th of December, 1861, as a Third Assistant Engineer. He served on board the *Chocura;* in the many battles in the Pamunky River; at Yorktown; Sabine Pass; and Calcasieu. After the war had ended he made a cruise on board the *Nyack,* in the Pacific, after which he became an instructor at the Naval Academy.

He was promoted to Second Assistant in 1863 and First Assistant in 1866, and resigned in 1872. He liked the service; liked the life and he resigned reluctantly; promotion being so slow, and every reorganization of the Navy reduced the engineers more than other corps, which discouraged Mr. Cooper. He accepted a lucrative position, offered by Capt. Eades, in charge of the construction of the St. Louis bridge: After that was finished he became Superintendent for both the Delaware and the Keystone Bridge Companies.

"When the first elevated railroads were built in New York City, Mr. Cooper was the assistant engineer in charge of construction. He was one of the five experts named by the President to figure on the Hudson River Bridge span and was consulting engineer for the Quebec Bridge and the New York Public Library. And the time the first cantilever bridge across

the St. Lawrence, at Quebec, collapsed and nearly 100 men lost their lives, it is said that the accident might have been avoided had a telegram sent by Mr. Cooper been received and heeded," says the *New York Sun* of 26th August, 1919.

Mr. Cooper was a member of the board of experts on the Manhattan Bridge plan in 1903. He was a lifelong member of the American Society of Civil Engineers, of the Mechanical Engineers, the Loyal Legion and N. J. Historic Society.

He was a robust man, rarely ever complaining, and his death came after a very brief illness of pneumonia, at his residence, 353 West 57th Street, New York. He was never married, which, he so often said, was a reason for remaining in the Navy. His life was a busy one, which, he said, brought content. He was of a genial disposition and had many friends.

G. W. B.

ASSOCIATION NOTES.

Council Meeting October 2, 1919.

Due to detachment from duty in Washington, Lieutenant Commander F. W. Sterling, U. S. N., Retired, and Captain of Engineers M. W. Torbet, U. S. C. G., tendered their resignations as Secretary-Treasurer and Member of Council, respectively, effective October 1, 1919. Commander Joseph S. Evans, U. S. N., was appointed Secretary-Treasurer and Captain of Engineers W. M. Prall, U. S. C. G., Member of Council for the remainder of the year 1919.

The matter of holding a banquet during 1919 was considered, and it was decided that conditions have not yet become sufficiently settled to warrant such action.

Annual Meeting.

The annual meeting of the Society for the nomination of officers for the year 1920, and for such other business as might be presented was held on Tuesday, October 7, 1919, pursuant to our By-Laws. The following were nominated:

For *President:*
Rear Admiral C. W. Dyson, U. S. N.

For *Secretary-Treasurer:*
Commander J. S. Evans, U. S. N.

For *Member of Council:*
Rear Admiral B. C. Bryan, U. S. N.
Captain Robert Stocker, C. C., U. S. N.
Commander S. C. Hooper, U. S. N.
Commander F. J. Cleary, U. S. N.
Commander Kenneth Whiting, U. S. N.
Commander H. T. Dyer, U. S. N.
Captain of Engineers W. M. Prall, U. S C. G.

It was voted to place the matter of holding a banquet during 1920 before the membership and provision has been made for voting on this question on the ballot for officers for 1920.

The Secretary-Treasurer desires to renew his request that members promptly notify him of changes in address.

The following members and associates have joined the Society since the publication of the last Journal:

MEMBERS.

Ames, Eugene, ex-Commissioned Officer, U. S. N.
Gary, C. B., Ensign, U. S. N.
Russell, George L., Lieut., U. S. N.

ASSOCIATES.

Abrams, Lawrence B., Washington, D. C.
Ball, Howard J., Washington, D. C.
Dias, Peter, Lieut. U. S. N. R. F.
Dodge, Owen, Colorado Springs, Colo.
Edmundson, W. A., Washington, D. C.
Harter, Isaac, Supt., Babcock & Wilcox Co., Bayonne, N. J.
Kessler, Raines, New York City.
London, William J. A., President, Steam Motors Co., Springfield, Mass.
Muir, Reginald L., New York City.
Williams, K. D., Washington, D. C.